Heating Ventilating Air Conditioning Guide 1938

HEATING VENTILATING
AIR CONDITIONING
GUIDE
1938

AN INSTRUMENT OF SERVICE PREPARED FOR THE PROFESSION—CONTAINING A

Technical Data Section

OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS—BASED ON THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND CO-OPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY

TOGETHER WITH A

Manufacturers' Catalog Data Section

CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT

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Complete Indexes

TO TECHNICAL AND CATALOG DATA SECTIONS

Vol. 16

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PREFACE TO THE 16th EDITION

THE general acceptance of the HEATING, VENTILATING, AIR CONDITIONING GUIDE as an authoritative source of scientific information on all engineering phases of the industry, has imposed each year since 1922 more and greater responsibilities on the Guide Publication Committee. Although extensive new material has been added to this edition, the ideals of the founders have been carefully preserved by presenting only authentic and current investigative results which have been tried and accepted in practice.

The sequence and arrangement of the various chapters have been changed to provide for a more logical grouping according to related subject matter, and a visual chapter index has been added. Probably, the greatest changes that have been made in the text are those relating to cooling phases of the industry. The chapters on Refrigerants and Air Drying Agents, Cooling Load, Central Systems for Cooling and Dehumidifying and Cooling and Dehumidification Methods have been entirely revised to bring them up-to-date and to correlate them more closely.

One entirely new chapter on Air Conditioning in the Treatment of Disease was introduced in this edition which outlines many of the application requirements found in the medical field. Some minor changes were made in the chapters on Heat Transmission Coefficients, Air Leakage, Heat and Fuel Utilization, Heating Boilers and Steam Heating Systems.

The chapters have been rewritten which deal with Air Pollution, Automatic Fuel Burning Equipment, Hot Water Heating Systems and Piping, Spray Equipment for Humidification and Dehumidification, Air Cleaning Devices, Railway Air Conditioning, Industrial Air Conditioning, Piping and Duct Insulation, Electrical Heating and Radiant Heating.

The Catalog Data Section of the GUIDE is receiving more recognition as each issue appears for the valuable product data contained therein. As usual, the manufacturers have cooperated in accomplishing the dual purpose of this part of the GUIDE; to provide authoritative and condensed catalog information for the Guide user, and to develop an effective and productive advertising medium for the manufacturer.

In offering the 16th edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE, the Committee wishes to acknowledge not only the editorial assistance unselfishly given by many whose aid is specifically accredited elsewhere, but also the valuable suggestions offered by the many readers and users of other editions. It is hoped that improvements and additions will be suggested to future Committees by readers of this volume, particularly, as perfection is not of human attainment.

Without the publication of previous editions, the issuance of the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938 in this fairly complete state in so short a time would be practically impossible. It is the Committee's hope that the same enthusiastic reception given to earlier volumes will be accorded this edition of 15,500 copies.

Albert Buenger, *Chairman*

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EDITORIAL ACKNOWLEDGMENT

FOR 16 years the HEATING, VENTILATING, AIR CONDITIONING GUIDE has been published by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and it retains its leadership as the authoritative reference volume of the profession and its allied industries. Its users are legion and they have adopted The Guide as their standard reference source because they recognize that the data are unbiased, up-to-date and readily usable. This has been made possible because of the willingness of hundreds of technical experts to contribute freely from their knowledge and practical experience for the advancement of the profession.

To the following individuals the Guide Publication Committee is profoundly grateful for their assistance in the production of the 1938 edition, and also to those former contributors who have previously prepared a firm foundation for the addition of new material.

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CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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HEATING VENTILATING AIR CONDITIONING

GUIDE

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Chapter 1

AIR, WATER AND STEAM

Dalton's Law, Temperatures, Air Properties, Humidity, Relative Humidity, Specific Humidity, Relation of Dew Point to Relative Humidity, Adiabatic Saturation of Air, Total Heat and Heat Content, Enthalpy, Psychrometric Chart, Properties of Water, Properties of Steam, Rate of Evaporation

AIR conditioning has for its objective the supplying and maintaining, in a room or other enclosure, of an atmosphere having a composition, temperature, humidity, and motion which will produce desired effects upon the occupants of the room or upon materials stored or handled in it.

Dry air is a mechanical mixture of gases composed, in percentage of volume, as follows[1]: nitrogen 78.03, oxygen 20.99, argon 0.94, carbon dioxide 0.03, and small amounts of hydrogen and other gases.

Atmospheric air at sea level is given in percentage by volume as: N_2 77.08, O_2 20.75, water vapor 1.2, A 0.93, CO_2 0.03 and H_2 0.01. The amount of water vapor varies greatly under different conditions and is frequently one of the most important constituents since it affects bodily comfort and greatly affects all kinds of hygroscopic materials.

DALTON'S LAW

A mixture of dry gases and water vapor, such as atmospheric air, obeys Dalton's Law of Partial Pressures; each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature had no other gas or vapor been present. If p = the observed pressure of the mixture

[1]*International Critical Tables.*

and p_1, p_2, p_3, etc. = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3, \text{ etc.} \tag{1}$$

TEMPERATURES

Air is said to be saturated at a given temperature when the water vapor mixed with the air is in the dry saturated condition or, what is the equivalent, when the space occupied by the mixture holds the maximum possible weight of water *vapor* at that temperature. If the water vapor mixed with the dry air is superheated, *i.e.*, if its temperature is above the temperature of saturation for the actual water vapor partial pressure, the air is not saturated.

The starting point of most applications of thermodynamic principles to air conditioning problems is the experimental determination of the dry-bulb and wet-bulb temperatures, and sometimes the barometric pressure.

The *dry-bulb temperature* of the air is the temperature indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air. The *wet-bulb temperature* is determined by a thermometer with its bulb encased in a fine mesh fabric bag moistened with clean water and whirled through the air until the thermometer assumes a steady temperature. This steady temperature is the result of a dynamic equilibrium between the rate at which heat is transferred from the air to the water on the bulb and the rate at which this heat is utilized in evaporating moisture from the bulb. The rate at which heat is transferred from the air to the water is substantially proportional to the wet-bulb depression $(t - t^1)$, while the rate of heat utilization in evaporation is proportional to the difference between the saturation pressure of the water at the wet-bulb temperature and the actual partial pressure of the water vapor in the air $(e^1 - e)$. Carrier's equation for this dynamic equilibrium is

$$\frac{e^1 - e}{t - t^1} = \frac{B - e^1}{2800 - 1.3t^1} \tag{2a}$$

In the form commonly used,

$$e = e^1 - \frac{(B - e^1)\,(t - t^1)}{2800 - 1.3t^1} \tag{2b}$$

where

e = actual partial pressure of water vapor in the air, inches of mercury.
e^1 = saturation pressure at wet-bulb temperature, inches of mercury.
B = barometric pressure, inches of mercury.
t = dry-bulb temperature, degrees Fahrenheit.
t^1 = wet-bulb temperature, degrees Fahrenheit.

Formula 2b may be used to determine the actual partial pressure of the water vapor in a dry air-water vapor mixture. Then, from Dalton's Law of Partial Pressures, Equation 1, it follows that the partial pressure of the dry air is $(B - e)$.

If a mixture of dry air and water vapor, initially unsaturated, be cooled

at constant pressure, the temperature at which condensation of the water vapor begins is called the *dew-point temperature.* Clearly the dew-point is the saturation temperature corresponding to the actual partial pressure, e, of the water vapor in the mixture.

AIR PROPERTIES

Density is variously defined as the mass per unit of volume, the weight per unit of volume, or the ratio of the mass, or weight, of a given volume of a substance to the mass, or weight, of an equal volume of some other substance such as water or air under standard conditions of temperature and pressure. The term *specific gravity* is more commonly used to express the latter relation but, when the gram is taken as the unit of mass and the cubic centimeter as the unit of volume, density and specific gravity have the same meaning. The term *specific density* is sometimes used to distinguish the weight in pounds per cubic foot; and as here used, *density* is the weight in pounds of one cubic foot of a substance.

The density of air decreases with increase in temperature when under constant pressure. The density of dry air at 70 F and under standard atmospheric pressure (29.921 in. of Hg) is approximately 0.075 lb (see Table 1), while that of a mixture of air and saturated water vapor at the same temperature and barometric pressure is only about 0.0742 lb. In the mixture the density of the dry air is 0.07307 and that of the vapor is 0.00115 lb (see Table 2).

In order to make comparisons of air volumes or velocities it is necessary to reduce the observations to a common pressure and temperature basis. The basic pressure is usually taken as 29.921 in. of Hg, but no basic temperature is universally recognized. Common temperatures for this purpose are 32 F, 60 F, 68 F, and 70 F. Since 70 F is the most commonly specified temperature to which rooms for human occupancy must be heated, it is usually understood, when no other temperature is specified, that 70 F is the basic temperature for measuring the volume or the velocity of air in heating and ventilating work.

The *specific volume* of air is the volume in cubic feet occupied by one pound of the air. Under constant pressure the specific volume varies inversely as the density and directly as the absolute temperature.

The *specific heat* of air is the number of Btu required to raise the temperature of 1 lb of air 1 F. Distinction should always be made between the instantaneous specific heat at any existent temperature and the mean specific heat, which is the average specific heat through a given temperature range. The mean specific heat is the value required in most calculations. The specific heats at constant pressure, C_p, and the specific heats, C_v, at constant volume are different. The specific heat at constant pressure is commonly used and it varies, under a pressure of one atmosphere, from a minimum at 32 F from which it increases with either increase or decrease of temperature. The value of 0.24, as the mean specific heat at constant pressure, is sufficiently accurate for use at ordinary temperatures. Values for instantaneous and mean specific heats are given in Table 3.

The *mean specific heat* of *water vapor* at constant pressure is taken as 0.45 for all general engineering computations.

TABLE 1. PROPERTIES OF DRY AIR[a]

Barometric Pressure 29.921 In. of Hg

Temperature Deg F	Weight Pounds per Cu Ft	Ratio of Volume to Volume at 70 F	Btu Absorbed by One Cu Ft Dry Air per Deg F	Cu Ft Dry Air Warmed One Deg per Btu
0	0.08633	0.8678	0.02077	48.15
10	0.08449	0.8867	0.02030	49.26
20	0.08273	0.9056	0.01986	50.35
30	0.08104	0.9245	0.01944	51.44
40	0.07942	0.9433	0.01905	52.49
50	0.07785	0.9624	0.01868	53.36
60	0.07636	0.9811	0.01832	54.44
70	0.07492	1.0000	0.01798	55.62
80	0.07353	1.0189	0.01765	56.66
90	0.07219	1.0378	0.01733	57.70
100	0.07090	1.0567	0.01702	58.75
110	0.06966	1.0755	0.01672	59.81
120	0.06845	1.0946	0.01643	60.86
130	0.06729	1.1133	0.01616	61.88
140	0.06617	1.1322	0.01589	62.93
150	0.06509	1.1510	0.01563	63.98
160	0.06403	1.1701	0.01538	65.02
180	0.06203	1.2078	0.01490	67.11
200	0.06015	1.2456	0.01446	69.24
220	0.05838	1.2832	0.01403	71.27
240	0.05671	1.3211	0.01364	73.31
260	0.05514	1.3587	0.01326	75.41
280	0.05365	1.3965	0.01291	77.46
300	0.05223	1.4344	0.01257	79.55
350	0.04901	1.5287	0.01181	84.67
400	0.04615	1.6234	0.01114	89.67
450	0.04362	1.7176	0.01054	94.87
500	0.04135	1.8119	0.01001	99.01
550	0.03930	1.9064	0.00953	104.93
600	0.03744	2.0011	0.00908	110.13
700	0.03422	2.1893	0.00833	120.05
800	0.03150	2.3784	0.00769	130.04
900	0.02911	2.5737	0.00713	140.25
1000	0.02718	2.7564	0.00668	149.70

[a]Compiled by W. H. Severns, based on the instantaneous specific heats of air. The values for the heats and for the cubic feet warmed one degree are for the temperatures stated and are not true over a temperature range of more than one degree above or below the temperatures stated.

TABLE 2. PROPERTIES OF SATURATED AIR[a]

Weights of Air, Vapor and Saturated Mixture of Air and Vapor at 29.921 In. of Hg

Temp. Deg F	Weight in a Cubic Foot of Mixture			Btu Absorbed by One Cubic Foot Sat. Air per Deg F	Cubic Feet Sat. Air Warmed One Deg per Btu	Specific Heat Btu per Pound of Mixture
	Weight of Dry Air Pounds	Weight of Vapor Pounds	Total Weight of Mixture Pounds			
0	0.08622	0.000068	0.08629	0.02078	48.12	0.2408
10	0.08431	0.000111	0.08442	0.02031	49.24	0.2406
20	0.08244	0.000177	0.08262	0.01987	50.33	0.2405
30	0.08060	0.000278	0.08088	0.01946	51.39	0.2406
40	0.07876	0.000409	0.07917	0.01908	52.41	0.2410
50	0.07692	0.000587	0.07751	0.01872	53.42	0.2415
60	0.07503	0.000828	0.07586	0.01838	54.41	0.2423
70	0.07307	0.001151	0.07422	0.01805	55.40	0.2432
80	0.07099	0.001578	0.07257	0.01775	56.34	0.2446
90	0.06877	0.002134	0.07090	0.01747	57.24	0.2464
100	0.06634	0.002851	0.06919	0.01721	58.11	0.2487
110	0.06361	0.003762	0.06737	0.01696	58.96	0.2517
120	0.06057	0.004912	0.06548	0.01675	59.70	0.2558
130	0.05712	0.006344	0.06346	0.01657	60.35	0.2611
140	0.05317	0.008116	0.06129	0.01642	60.91	0.2679
150	0.04863	0.010284	0.05891	0.01630	61.35	0.2767
160	0.04339	0.012919	0.05631	0.01624	61.58	0.2884
170	0.03733	0.016092	0.05342	0.01621	61.69	0.3034
180	0.03033	0.019888	0.05022	0.01624	61.58	0.3234
190	0.02228	0.024384	0.04666	0.01633	61.24	0.3500
200	0.01298	0.029700	0.04268	0.01649	60.64	0.3864
210	0.00230	0.035932	0.03616	0.01672	59.81	0.4624
212	0.00000	0.037286	0.03729	0.01818	55.01	0.4875

[a]Compiled by W. H. Severns, based on the instantaneous specific heats of air.

TABLE 3. SPECIFIC HEATS OF DRY AIR[a]

Constant Barometric Pressure of 29.921 In. of Hg

Temperature Deg F	Instantaneous or True Specific Heat	Temperature Range Deg F	Mean Specific Heat
−301.0	0.2520	32 to 212	0.2401
−108.4	0.2430	32 to 392	0.2411
32.0	0.2399	32 to 752	0.2420
212.0	0.2403	32 to 1112	0.2430
392.0	0.2413	....	....
752.0	0.2430	....	....
1112.0	0.2470	....	....

[a]Compiled by W. H. Severns, based on data given in the *International Critical Tables*.

TABLE 4. WEIGHTS OF SATURATED AND PARTLY SATURATED AIR FOR VARIOUS BAROMETRIC AND HYGROMETRIC CONDITIONS[a, b]

Pounds per Cubic Foot

Dry-Bulb Temperature Deg F	Barometric Pressure—Inches															Approx. Average Increase in Weight per Deg Wet-Bulb Depression
	26			27			28			29			30			
	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt per Deg Inc Dry-Bulb	Incr's Wt per 0.1″ Rise in Bar.	
0	0.07500	0.00016	0.00029	0.07788	0.00016	0.00029	0.08077	0.00017	0.00029	0.08365	0.00018	0.00029	0.08654	0.00019	0.00029	0.000015
10	0.07338	0.00016	0.00028	0.07620	0.00016	0.00028	0.07903	0.00017	0.00028	0.08185	0.00018	0.00028	0.08468	0.00018	0.00028	0.000016
20	0.07180	0.00016	0.00028	0.07456	0.00016	0.00028	0.07733	0.00017	0.00028	0.08009	0.00018	0.00028	0.08286	0.00018	0.00028	0.000017
30	0.07027	0.00015	0.00027	0.07297	0.00016	0.00027	0.07569	0.00016	0.00027	0.07839	0.00017	0.00027	0.08110	0.00017	0.00027	0.000019
40	0.06879	0.00015	0.00026	0.07143	0.00015	0.00027	0.07409	0.00016	0.00027	0.07675	0.00016	0.00027	0.07942	0.00017	0.00027	0.000021
50	0.06732	0.00015	0.00026	0.06992	0.00015	0.00026	0.07252	0.00016	0.00026	0.07512	0.00016	0.00026	0.07773	0.00016	0.00026	0.000023
60	0.06588	0.00015	0.00026	0.06843	0.00015	0.00026	0.07098	0.00015	0.00026	0.07353	0.00016	0.00026	0.07609	0.00016	0.00026	0.000026
70	0.06442	0.00015	0.00025	0.06692	0.00015	0.00025	0.06943	0.00015	0.00025	0.07193	0.00016	0.00025	0.07440	0.00016	0.00025	0.000029
80	0.06297	0.00015	0.00025	0.06542	0.00015	0.00025	0.06789	0.00015	0.00025	0.07034	0.00015	0.00025	0.07280	0.00016	0.00025	0.000034
90	0.06146	0.00015	0.00024	0.06388	0.00016	0.00024	0.06629	0.00016	0.00024	0.06870	0.00016	0.00024	0.07112	0.00017	0.00024	0.000039
100	0.05991	0.00016	0.00024	0.06228	0.00016	0.00024	0.06465	0.00016	0.00024	0.06703	0.00017	0.00024	0.06939	0.00018	0.00024	0.000044
110	0.05828	0.00016	0.00023	0.06060	0.00017	0.00023	0.06293	0.00017	0.00023	0.06526	0.00018	0.00023	0.06759	0.00019	0.00023	0.000051
120	0.05653	0.00018	0.00023	0.05882	0.00018	0.00023	0.06111	0.00018	0.00023	0.06339	0.00019	0.00023	0.06569	0.00020	0.00023	0.000059
130	0.05467	0.00019	0.00023	0.05692	0.00019	0.00023	0.05917	0.00019	0.00023	0.06142	0.00020	0.00023	0.06367	0.00022	0.00023	0.000068
140	0.05262	0.00021	0.00022	0.05483	0.00021	0.00022	0.05704	0.00021	0.00022	0.05925	0.00022	0.00022	0.06147	0.00024	0.00022	0.000078
150	0.05036	0.00023	0.00022	0.05253	0.00023	0.00022	0.05471	0.00023	0.00022	0.05689	0.00024	0.00022	0.05906	0.00026	0.00022	0.000090
160	0.04788	0.00025	0.00022	0.05001	0.00025	0.00022	0.05216	0.00026	0.00021	0.05430	0.00026	0.00021	0.05644	0.00029	0.00021	0.000103
170	0.04509	0.00028	0.00021	0.04720	0.00028	0.00021	0.04931	0.00029	0.00021	0.05141	0.00031	0.00021	0.05352	0.00033	0.00021	0.000118
180	0.04197	0.00031	0.00021	0.04404	0.00031	0.00021	0.04611	0.00032	0.00021	0.04818	0.00034	0.00021	0.05026	0.00036	0.00021	0.000134
190	0.03845	0.00035	0.00021	0.04049	0.00036	0.00021	0.04253	0.00036	0.00021	0.04457	0.00037	0.00021	0.04662	0.00038	0.00021	0.000153
200	0.03449	0.00040	0.00020	0.03650	0.00040	0.00020	0.03851	0.00040	0.00020	0.04052	0.00041	0.00020	0.04254	0.00041	0.00020	0.000173

[a]From *Fan Engineering*.

[b]A convenient and accurate chart for quickly determining the weight of air under any condition of dry-bulb, wet-bulb, and pressure is *A Chart for Determining the Weight of Moist Air in Pounds per Cubic Foot*, by John E. Younger. Published in *Mechanical Engineering*, June, 1925.

Table 4 is intended to aid in determining the density of moist air, taking into account its temperature, pressure, and moisture content.

Example 1. To show the use of Table 4: Given air at 83 F dry-bulb and 68 F wet-bulb (or a depression of 15 deg) with a barometric pressure of 29.40 in. of mercury. What will be the weight of this air in pounds per cubic foot?

Solution. From Table 4 the weight of saturated air at 80 F and 29.00 in. barometer is found to be 0.07034 lb per cubic foot. There is a decrease of 0.00015 lb per degree dry-bulb temperature above 80 F. There is an increase of 0.00025 lb for each 0.1 in. above 29.00 in. From the last column of Table 4 it is found that there is an increase of approximately 0.000035 lb per degree wet-bulb depression when the dry-bulb is 83 F. Tabulating the items:

0.07034 = weight of saturated air at 80 F and 29.00 bar.
− 0.00045 = decrement for 3 deg dry-bulb, 3 × 0.00015.
+ 0.00100 = increment for 0.4 in. bar., 4 × 0.00025.
+ 0.00053 = increment for 15 deg wet-bulb depression, 15 × 0.000035.

0.07142 = weight in pounds per cubic foot of air at 83 F dry-bulb, 68 F wet-bulb, 29.40 in. bar.

It is usual to assume that dry air, moist air, and the water vapor in the air follow the laws of perfect gases. This assumption while not absolutely true, especially with saturated vapor at temperatures much above 140 F, is sufficiently accurate for practical purposes and it greatly simplifies computations.

Boyle's Law refers to the relation between the pressure and volume of a gas, and may be stated as follows: *With temperature constant, the volume of a given weight of gas varies inversely as its absolute pressure.* Hence, if P_1 and P_2 represent the initial and final absolute pressures, and V_1 and V_2 represent corresponding volumes of the same mass, say one pound of gas, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$, or $P_1 V_1 = P_2 V_2$, but since $P_1 V_1$ for any given case is a definite constant quantity, it follows that the product of the absolute pressure and volume of a gas is a constant, or $PV = C$, when T is kept constant. Any change in the pressure and volume of a gas at constant temperature is called an *isothermal change.*

Charles' Law refers to the relation among pressure, volume, and temperature of a gas and may be stated as follows: *The volume of a given weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume.* Hence, when heat is added at constant volume, V_c, the resulting equation is $\frac{P_2}{P_1} = \frac{T_2}{T_1}$, or, for the same temperature range at constant pressure, P_c, the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general, for any weight of gas, W, since volume is proportional to weight, the relation among P, V, and T is

$$PV = WRT \tag{3}$$

where

P = the absolute pressure of the gas, pounds per square foot.
V = the volume of the weight W, cubic feet.

W = the weight of the gas, pounds.
R = a constant depending on the nature of the gas. The average value of R for air is 53.34.
T = the absolute temperature, degrees Fahrenheit.

This is the characteristic equation for a perfect gas, and while no gases are perfect in this sense, they conform so nearly that Equation 3 will apply to most engineering computations.

HUMIDITY

Humidity is the water vapor mixed with dry air in the atmosphere. *Absolute humidity* has a multiplicity of meanings, but usually the term refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. With this meaning, absolute humidity is nothing but the actual density of the water vapor in the mixture and might better be so called. A study of Keenan's Steam Tables[2] indicates that water vapor, either saturated or super-heated, at partial pressures lower than 4 in. of mercury may be treated as a gas with a gas constant R of 1.21 in the characteristic equation of the gas $pV = wR\ (t + 460)$. Within such limits, the density (d) of water vapor is

$$d = \frac{w}{V} = \frac{e}{1.21\ (t + 460)} \text{ (pounds per cubic foot)} \qquad (4a)$$

$$= \frac{5785\ e}{t + 460} \text{ (grains per cubic foot)} \qquad (4b)$$

where

e = actual partial pressure of vapor, inches of mercury.
t = dry-bulb temperature, degrees Fahrenheit.

Specific Humidity

It simplifies many problems which deal with mixtures of dry air and water vapor to express the weight or the mass of the vapor in terms of the weight or the mass of dry air. If the weight of the water vapor in a mixture be divided by the weight of the dry air, and the weight of dry air be made unity, we have an expression of the weight of water vapor carried by a unit weight of dry air. This relation has no generally accepted name. It has been variously called: mixing ratio, proportionate humidity, mass or density ratio, absolute humidity, and specific humidity. Of all these terms *specific humidity* is the most suggestive of the meaning which it is desired to express and it has found considerable use in this sense even though it is defined in *International Critical Tables* as the ratio of the mass of vapor to the total mass. It will be understood here that *specific humidity* refers to the weight of water vapor carried by one pound of dry air.

The gas constant for dry air, when the partial pressure of the air is expressed in inches of Hg, is 0.753; so that the specific humidity, if represented by W, is

[2]Published by *American Society of Mechanical Engineers*, see abstract in Table 8.

$$W = \frac{e}{1.21\ (t + 460)} \div \frac{B\text{-}e}{0.753\ (t + 460)}$$

$$= 0.622 \left(\frac{e}{B\text{-}e}\right) \text{ (pounds)} \tag{5a}$$

$$= 4354 \left(\frac{e}{B\text{-}e}\right) \text{ (grains)} \tag{5b}$$

where

e = actual partial pressure of vapor, inches of mercury.
B = total pressure of mixture (barometric pressure), inches of mercury.

Relative Humidity

Relative humidity (Φ) is either the ratio of the actual partial pressure, e, of the water vapor in the air to the saturation pressure, e_t, at the dry-bulb temperature, or the ratio of the actual density, d, of the vapor to the density of saturated vapor, d_t, at the dry-bulb temperature. That is:

$$\Phi = \frac{e}{e_t} = \frac{d}{d_t} \tag{6}$$

The relative humidity of a given mixture at a given temperature is not the same as the specific humidity, W, of the mixture divided by the specific humidity, W_t, of saturated vapor at the same temperature, for from Equations 5a and 6

$$\frac{W}{W_t} = 0.622 \left(\frac{\Phi\, e_t}{B - \Phi\, e_t}\right) \div 0.622 \left(\frac{e_t}{B\text{-}e_t}\right) = \frac{\Phi\ (B\text{-}e_t)}{B - \Phi\, e_t} \tag{7}$$

The specific humidity of an unsaturated air-vapor mixture cannot, therefore, be accurately found by multiplying the specific humidity of saturated vapor by its relative humidity; although the error is usually small especially when the relative humidity is high.

With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and dew-point temperatures are equal. With a relative humidity less than 100 per cent, the dry-bulb exceeds the wet-bulb, and the wet-bulb exceeds the dew-point temperature.

RELATION OF DEW POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew-point and the relative humidity and this is found most useful in air conditioning work. This relationship is, that for a fixed relative humidity there is substantially a constant difference between the dew-point and the dry-bulb temperature over a considerable temperature range. Table 5, giving the dry-bulb and dew-point temperatures and the dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

TABLE 5. TEMPERATURES FOR 50 PER CENT RELATIVE HUMIDITY

Dry-bulb temperature	65.0	70.0	75.0	80.0	85.0	90.0
Dew-point temperature	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew-point and dry-bulb temperature	19.2	19.5	19.75	20.25	20.75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_1}{20}}} \tag{8}$$

where

t_1 = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills and tobacco factories, where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and the dew-point temperature, to control the relative humidity in the room.

Table 6 gives, for different temperatures, the density of saturated vapor, d_t, the weight of saturated vapor mixed with 1 lb of dry air, W_t, (at a relative humidity of 100 per cent and a barometric pressure, B, of 29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 lb of dry air (at a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The preceding equations or the data from Table 6 may be conveniently used in solving the following typical problems:

Example 2. Humidifying and Heating. Air is to be maintained at 70 F with a relative humidity of 40 per cent (Φ = 0.4) when the outside air is at 0 F and 70 per cent relative humidity (Φ = 0.7) and a barometric pressure, B, of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Equation 5a and Table 6,

$$W_1 = 0.622\left(\frac{0.7 \times 0.03773}{29.92 - 0.0264}\right) = 0.000548 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622\left(\frac{0.4 \times 0.7386}{29.92 - 0.295}\right) = 0.00618 \text{ lb per pound of dry air.}$$

The water vapor added per pound of dry air must be ($W_2 - W_1$) or 0.005632 lb. By inspection of Table 6, W_t = 0.00618 at 44.5 F, so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 6 is

W_1 = 0.7 × 0.0007852 = 0.00054964 lb per pound of dry air.
W_2 = 0.4 × 0.01574 = 0.006296 lb per pound of dry air.

The water vapor added per pound of dry air is approximately 0.00574636 lb and the dew-point temperature is approximately 45 F. The degree of approximation is evident.

Example 3. Dehumidifying and Cooling. Air with a dry-bulb temperature of 84 F, a wet-bulb of 70 F, or a relative humidity of 50 per cent ($\Phi = 0.5$), and a barometric pressure, B, of 29.92 in. of mercury is to be cooled to 54 F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Equation 5a and Table 6,

$$W_1 = 0.622 \left(\frac{0.5 \times 1.1752}{29.92 - 0.5876}\right) = 0.01248 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0.42003}{29.92 - 0.42003}\right) = 0.00887 \text{ lb per pound of dry air.}$$

Since $W_1 = W_t$ when $t = 63.4$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is $(W_1 - W_2)$ or 0.00361 lb per pound of dry air.

An approximate result is

$W_1 = 0.5 \times 0.02543 = 0.012715$ lb per pound of dry air.
$W_2 = 1 \times 0.008856 = 0.008856$ lb per pound of dry air, since the exit air is saturated.

Since $W_1 = W_t$ at $t = 64$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.003859 lb per pound of dry air. The degree of approximation is again evident.

ADIABATIC SATURATION OF AIR

The process of adiabatic saturation of air is of considerable importance in air conditioning. Suppose that 1 lb of dry air, initially unsaturated but carrying W lb of water vapor with a dry-bulb temperature, t, and a wet-bulb temperature, t', be made to pass through a tunnel containing an exposed water surface. Further assume the tunnel to be completely insulated, thermally, so that the only heat transfer possible is that between the air and water. As the air passes over the water surface, it will gradually pick up water vapor and will approach saturation *at the initial wet-bulb temperature of the air*, if the water be supplied at this wet-bulb temperature. During the process of adiabatic saturation, then, the dry-bulb temperature of the air drops to the wet-bulb temperature as a limit, the wet-bulb temperature remains substantially constant, and the weight of water vapor associated with each pound of dry air increases to $W_{t'}$, as a limit, *where* $W_{t'}$ is the weight of saturated vapor per pound of dry air for saturation at the wet-bulb temperature.

Example 4. If air with a dry-bulb of 85 F and a wet-bulb of 70 F be saturated adiabatically by spraying with recirculated water, what will be the final temperature and the vapor content of the air?

Solution. The final temperature will be equal to the initial wet-bulb temperature or 70 F, and since the air is saturated at this temperature, from Table 6, $W = 0.01574$ lb per pound of dry air.

In the adiabatic saturation process, since the heat given up by the dry air and associated vapor in cooling to the wet-bulb temperature is utilized in evaporation of water at the wet-bulb temperature, W. H. Carrier has pointed out[3] that the equation for the process of *adiabatic saturation*, and hence for a process of *constant wet-bulb temperature*, is:

[3]Rational Psychrometric Formulae, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 33, 1911, p 1005.)

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I)

Temp., F	Pressure of Saturated Vapor × 10⁻⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds × 10⁻⁵	Grains	Pounds × 10⁻⁵	Grains					
−130	0.276	0.1356	0.000693	0.000049	0.005738	0.00040	8.31	8.31	−31.71	1000.7	−31.71
−129	.306	.1503	.000766	.000054	.006362	.00045	8.33	8.33	−31.46	1001.2	−31.46
−128	.338	.1660	.000843	.000059	.007027	.00049	8.36	8.36	−31.21	1001.6	−31.21
−127	.373	.1832	.000928	.000065	.007755	.00054	8.38	8.38	−30.96	1002.1	−30.96
−126	.411	.2019	.001019	.000071	.008545	.00060	8.41	8.41	−30.71	1002.5	−30.71
−125	0.455	0.2235	0.001125	0.000079	0.009459	0.00066	8.43	8.43	−30.46	1003.0	−30.46
−124	.499	.2451	.001230	.000086	.01037	.00073	8.46	8.46	−30.21	1003.4	−30.21
−123	.542	.2662	.001332	.000093	.01127	.00079	8.48	8.48	−29.96	1003.9	−29.96
−122	.604	.2967	.001480	.00010	.01256	.00088	8.51	8.51	−29.72	1004.3	−29.72
−121	.669	.3286	.001635	.00012	.01391	.00097	8.53	8.53	−29.47	1004.8	−29.47
−120	0.735	0.3610	0.001791	0.00013	0.01528	0.00107	8.56	8.56	−29.22	1005.2	−29.22
−119	.805	.3954	.001956	.00014	.01674	.00117	8.58	8.58	−28.97	1005.7	−28.97
−118	.892	.4382	.002161	.00015	.01854	.00130	8.61	8.61	−28.72	1006.1	−28.72
−117	.989	.4858	.002388	.00017	.02056	.00144	8.63	8.63	−28.47	1006.6	−28.47
−116	1.098	.5393	.002644	.00019	.02283	.00160	8.66	8.66	−28.23	1007.0	−28.23
−115	1.208	0.5934	0.002900	0.00020	0.02511	0.00176	8.68	8.68	−27.98	1007.5	−27.98
−114	1.317	.6469	.003153	.00022	.02738	.00192	8.71	8.71	−27.73	1007.9	−27.73
−113	1.444	.7093	.003447	.00024	.03002	.00210	8.73	8.73	−27.48	1008.4	−27.48
−112	1.575	.7736	.003749	.00026	.03274	.00229	8.76	8.76	−27.23	1008.8	−27.23
−111	1.728	.8488	.004101	.00029	.03593	.00252	8.78	8.78	−26.99	1009.3	−26.99
−110	1.889	0.9279	0.004471	0.00031	0.03927	0.00275	8.81	8.81	−26.74	1009.7	−26.74
−109	2.087	1.0251	.004925	.00035	.04339	.00304	8.83	8.83	−26.49	1010.2	−26.49
−108	2.292	1.1258	.005393	.00038	.04765	.00334	8.86	8.86	−26.24	1010.6	−26.24
−107	2.511	1.2334	.005892	.00041	.05220	.00365	8.89	8.89	−26.00	1011.1	−26.00
−106	2.742	1.3469	.006415	.00045	.05701	.00399	8.91	8.91	−25.75	1011.5	−25.75
−105	2.983	1.4652	0.006960	0.00049	0.06202	0.00434	8.94	8.94	−25.50	1012.0	−25.50
−104	3.258	1.6003	.007580	.00053	.06773	.00474	8.96	8.96	−25.26	1012.4	−25.26
−103	3.543	1.7403	.008219	.00058	.07366	.00516	8.99	8.99	−25.01	1012.9	−25.01
−102	3.872	1.9019	.008958	.00063	.08050	.00564	9.01	9.01	−24.76	1013.3	−24.76
−101	4.213	2.0694	.009719	.00068	.08759	.00613	9.04	9.04	−24.51	1013.8	−24.51
−100	4.607	2.2630	0.010599	0.00074	0.09578	0.00705	9.06	9.06	−24.27	1014.2	−24.27
−99	5.018	2.4637	.011512	.00081	.1043	.00730	9.09	9.09	−24.02	1014.7	−24.02
−98	5.455	2.6784	.012480	.00087	.1134	.00794	9.11	9.11	−23.78	1015.1	−23.78
−97	5.946	2.9196	.013566	.00095	.1236	.00865	9.14	9.14	−23.53	1015.6	−23.53
−96	6.470	3.1781	.014721	.00103	.1345	.00942	9.16	9.16	−23.28	1016.0	−23.28

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Table*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONTINUED)

Temp., F	Pressure of Saturated Vapor × 10⁻⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
			per Cu Ft		per lb of Dry Air						
	In. of Hg	Lb per Sq In.	Pounds × 10^{-5}	Grains	Pounds × 10^{-5}	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
−95	7.047	3.4604	0.015990	0.00112	0.1465	0.01026	9.19	9.19	−23.04	1016.5	−23.04
−94	7.638	3.7507	.017284	.00121	.1588	.01112	9.21	9.21	−22.79	1016.9	−22.79
−93	8.316	4.0837	.018767	.00131	.1729	.01210	9.23	9.23	−22.55	1017.4	−22.55
−92	9.017	4.4281	.020292	.00142	.1875	.01312	9.26	9.26	−22.30	1017.8	−22.30
−91	9.806	4.8156	.022009	.00154	.2039	.01427	9.29	9.29	−22.05	1018.3	−22.05
−90	10.64	5.2264	0.023817	0.00167	0.2212	0.01548	9.31	9.31	−21.81	1018.7	−21.81
−89	11.53	5.6635	.025738	.00180	.2397	.01678	9.34	9.34	−21.56	1019.2	−21.56
−88	12.51	6.1449	.027851	.00195	.2601	.01821	9.36	9.36	−21.32	1019.6	−21.32
−87	13.53	6.6459	.030041	.00210	.2813	.01969	9.39	9.39	−21.07	1020.1	−21.07
−86	14.69	7.2157	.032530	.00228	.3054	.02138	9.41	9.41	−20.83	1020.5	−20.83
−85	15.87	7.7953	0.035049	0.00245	0.3299	0.02309	9.44	9.44	−20.58	1021.0	−20.58
−84	17.20	8.4486	.037885	.00265	.3576	.02503	9.46	9.46	−20.34	1021.4	−20.34
−83	18.58	9.1265	.040817	.00286	.3863	.02704	9.49	9.49	−20.09	1021.9	−20.09
−82	20.10	9.8731	.044037	.00308	.4179	.02925	9.51	9.51	−19.84	1022.3	−19.84
−81	21.72	10.669	.047463	.00332	.4516	.03161	9.54	9.54	−19.60	1022.8	−19.60
−80	23.47	11.517	0.051151	0.00358	0.4879	0.03415	9.56	9.56	−19.36	1023.2	−19.36
−79	25.34	12.436	.055082	.00386	.5268	.03688	9.59	9.59	−19.11	1023.7	−19.10
−78	27.29	13.394	.059165	.00414	.5674	.03972	9.61	9.61	−18.87	1024.1	−18.86
−77	29.52	14.489	.063831	.00447	.6137	.04296	9.64	9.64	−18.62	1024.6	−18.61
−76	31.81	15.614	.068605	.00480	.6613	.04629	9.66	9.66	−18.38	1025.0	−18.37
−75	34.37	16.883	0.073933	0.00518	0.7146	0.05002	9.69	9.69	−18.13	1025.5	−18.12
−74	37.01	18.179	.079405	.00556	.7694	.05386	9.72	9.72	−17.89	1025.9	−17.88
−73	39.96	19.628	.085510	.00599	.8308	.05816	9.74	9.74	−17.64	1026.4	−17.63
−72	43.04	21.141	.091865	.00643	.8948	.06264	9.77	9.77	−17.40	1026.8	−17.39
−71	46.33	22.757	.098632	.00690	.9632	.06742	9.79	9.79	−17.16	1027.3	−17.15
−70	49.87	24.496	0.10590	0.00741	1.037	0.07259	9.82	9.82	−16.91	1027.7	−16.90
−69	53.59	26.323	.11350	.00795	1.114	.07798	9.84	9.84	−16.67	1028.2	−16.66
−68	57.65	28.318	.12179	.00853	1.199	.08393	9.87	9.87	−16.42	1028.6	−16.41
−67	61.81	30.361	.13024	.00911	1.285	.08995	9.89	9.89	−16.18	1029.1	−16.17
−66	66.41	32.621	.13959	.00977	1.381	.09667	9.92	9.92	−15.94	1029.5	−15.93
−65	71.17	34.959	0.14922	0.01044	1.480	0.10360	9.94	9.94	−15.69	1030.0	−15.67
−64	76.64	37.646	.16028	.01122	1.593	.11151	9.97	9.97	−15.45	1030.4	−15.43
−63	82.28	40.416	.17164	.01201	1.711	.11977	9.99	9.99	−15.21	1030.9	−15.19
−62	88.19	43.319	.18350	.01285	1.833	.12831	10.02	10.02	−14.96	1031.3	−14.94
−61	94.62	46.477	.19638	.01375	1.967	.13769	10.04	10.04	−14.72	1031.8	−14.70

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONTINUED)

Temp., F	Pressure of Saturated Vapor $\times 10^{-5}$		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft: Pounds $\times 10^{-5}$	per Cu Ft: Grains	per lb of Dry Air: Pounds $\times 10^{-5}$	per lb of Dry Air: Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
−60	101.4	49.808	0.20993	0.01470	2.108	0.14756	10.07	10.07	−14.48	1032.2	−14.46
−59	108.8	53.443	.22469	.01573	2.262	.15834	10.09	10.09	−14.23	1032.7	−14.21
−58	116.3	57.127	.23058	.01677	2.418	.16926	10.12	10.12	−13.99	1033.1	−13.97
−57	124.8	61.302	.25645	.01795	2.595	.18165	10.14	10.14	−13.75	1033.6	−13.72
−56	133.4	65.526	.27344	.01914	2.773	.19411	10.17	10.17	−13.50	1034.0	−13.47
−55	143.0	70.242	0.29239	0.02047	2.973	0.20811	10.19	10.19	−13.26	1034.5	−13.23
−54	153.0	75.154	.31207	.02184	3.181	.22267	10.22	10.22	−13.02	1034.9	−12.99
−53	163.5	80.311	.33267	.02329	3.399	.23793	10.24	10.24	−12.78	1035.4	−12.74
−52	174.9	85.911	.35499	.02485	3.636	.25452	10.27	10.27	−12.53	1035.8	−12.49
−51	187.0	91.854	.37862	.02650	3.888	.27216	10.29	10.29	−12.29	1036.3	−12.25
−50	199.9	98.101	0.40376	0.02826	4.156	0.29092	10.32	10.32	−12.05	1036.7	−12.01
−49	213.0	104.63	.42917	.03004	4.428	.30996	10.34	10.34	−11.81	1037.2	−11.76
−48	227.9	111.94	.45808	.03207	4.738	.33166	10.37	10.37	−11.57	1037.6	−11.52
−47	243.1	119.41	.48744	.03412	5.054	.35378	10.40	10.40	−11.32	1038.1	−11.27
−46	259.5	127.47	.51905	.03633	5.395	.37765	10.42	10.42	−11.08	1038.5	−11.02
−45	276.7	135.92	0.55213	0.03865	5.753	0.40271	10.45	10.45	−10.84	1039.0	−10.78
−44	295.0	144.90	.58722	.04111	6.133	.42931	10.47	10.47	−10.60	1039.4	−10.54
−43	314.7	154.58	.62493	.04375	6.543	.45801	10.50	10.50	−10.35	1039.9	−10.28
−42	335.3	164.70	.66426	.04650	6.971	.48797	10.52	10.52	−10.11	1040.3	−10.04
−41	357.6	175.65	.70672	.04947	7.435	.52045	10.55	10.55	−9.872	1040.8	−9.795
−40	380.3	186.80	0.7498	0.05249	7.907	0.55349	10.57	10.57	−9.629	1041.2	−9.547
−39	405.5	199.18	.7976	.05583	8.431	.59017	10.60	10.60	−9.388	1041.7	−9.300
−38	431.2	211.81	.8461	.05922	8.965	.62755	10.62	10.62	−9.146	1042.1	−9.053
−37	459.2	225.56	.8989	.06292	9.548	.66836	10.65	10.65	−8.905	1042.6	−8.805
−36	488.4	239.90	.9538	.06677	10.16	.71120	10.67	10.67	−8.663	1043.0	−8.557
−35	519.5	255.18	1.0122	0.07085	10.80	0.75600	10.69	10.69	−8.422	1043.5	−8.309
−34	552.4	271.34	1.0738	.07517	11.49	.80430	10.72	10.72	−8.180	1043.9	−8.060
−33	586.5	288.09	1.1374	.07962	12.20	.85400	10.75	10.75	−7.939	1044.4	−7.812
−32	623.7	306.36	1.2067	.08447	12.97	.90790	10.77	10.77	−7.698	1044.8	−7.562
−31	661.8	325.08	1.2774	.08942	13.76	.96320	10.80	10.80	−7.457	1045.3	−7.313
−30	701.0	344.33	1.3499	0.09449	14.58	1.0206	10.82	10.82	−7.216	1045.7	−7.064
−29	742.2	364.57	1.4260	.09982	15.43	1.0801	10.85	10.85	−6.975	1046.2	−6.814
−28	791.2	388.64	1.5166	.10616	16.45	1.1515	10.87	10.87	−6.734	1046.6	−6.562
−27	841.0	413.10	1.6083	.11258	17.49	1.2243	10.90	10.90	−6.493	1047.1	−6.310
−26	892.1	438.20	1.7021	.11914	18.55	1.2985	10.92	10.92	−6.251	1047.5	−6.057

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONCLUDED)

TEMP., F	PRESSURE OF SATURATED VAPOR × 10^{-5}		WEIGHT OF SATURATED VAPOR				VOLUME IN CU FT BAROMETER, 29.92 IN. HG		HEAT CONTENT PER LB		
	In. of Hg	Lb per Sq In.	per Cu Ft: Pounds × 10^{-5}	per Cu Ft: Grains	per lb of Dry Air: Pounds × 10^{-5}	per lb of Dry Air: Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
−25	946.4	464.87	1.8016	0.12611	19.68	1.3776	10.95	10.95	−6.011	1048.0	−5.805
−24	1003.	492.67	1.9049	.13334	20.86	1.4602	10.97	10.97	−5.770	1048.4	−5.551
−23	1064.	522.64	2.0162	.14113	22.13	1.5491	11.00	11.00	−5.529	1048.9	−5.297
−22	1126.	553.09	2.1287	.14901	23.42	1.6394	11.02	11.02	−5.288	1049.3	−5.042
−21	1192.	585.51	2.2484	.15739	24.79	1.7353	11.05	11.05	−5.047	1049.8	−4.787
−20	1262.0	619.89	2.3750	0.16625	26.25	1.8375	11.07	11.07	−4.807	1050.2	−4.531
−19	1337.	656.73	2.5105	.17574	27.81	1.9467	11.10	11.10	−4.566	1050.7	−4.274
−18	1416.	695.54	2.6527	.18569	29.45	2.0615	11.13	11.13	−4.325	1051.1	−4.015
−17	1496.	734.84	2.7963	.19574	31.12	2.1784	11.15	11.15	−4.085	1051.6	−3.758
−16	1584.	778.06	2.9542	.20679	32.95	2.3065	11.18	11.18	−3.844	1052.0	−3.497
−15	1675.0	822.76	3.1168	0.21818	34.84	2.4388	11.20	11.21	−3.604	1052.5	−3.237
−14	1772.	870.41	3.2899	.23029	36.86	2.5802	11.23	11.24	−3.363	1052.9	−2.975
−13	1874.	920.51	3.4714	.24300	38.98	2.7286	11.25	11.26	−3.123	1053.4	−2.712
−12	1980.	972.58	3.6596	.25617	41.19	2.8833	11.28	11.29	−2.883	1053.8	−2.449
−11	2093.	1028.1	3.8599	.27019	43.54	3.0478	11.30	11.31	−2.642	1054.3	−2.183
−10	2210.0	1085.6	4.0666	0.28466	45.98	3.2186	11.33	11.34	−2.402	1054.7	−1.917
−9	2335.	1147.0	4.2871	.30009	48.58	3.4006	11.35	11.36	−2.162	1055.2	−1.649
−8	2463.	1209.8	4.5120	.31584	51.25	3.5875	11.38	11.39	−1.921	1055.6	−1.380
−7	2502.	1229.0	4.5734	.32014	52.06	3.6442	11.40	11.41	−1.681	1056.1	−1.131
−6	2745.	1348.3	5.0066	.35046	57.12	3.9984	11.43	11.44	−1.441	1056.5	−0.8375
−5	2898.0	1423.5	5.2738	0.36917	60.30	4.2210	11.45	11.46	−1.201	1057.0	−0.5636
−4	3055.	1500.6	5.5473	.38831	63.57	4.4499	11.48	11.49	−0.9604	1057.4	−0.2882
−3	3222.	1582.6	5.8379	.40865	67.05	4.6935	11.50	11.51	−0.7203	1057.9	−0.01098
−2	3397.	1668.6	6.1414	.42990	70.69	4.9483	11.53	11.54	−0.4802	1058.3	+0.2679
−1	3580.	1758.5	6.4583	.45208	74.50	5.2150	11.55	11.57	−0.2401	1058.8	+0.5487
0	3773.0	1853.3	6.7914	0.47500	78.52	5.5000	11.58	11.59	0	1059.2	+0.8317

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft Pounds	per Cu Ft Grains	per lb of Dry Air Pounds	per lb of Dry Air Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
0	0.03773	0.01853	0.000067914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317
1	.03975	.01963	.000071395	.500	.0008275	5.79	11.60	11.62	.2401	1059.7	1.117
2	.04186	.02056	.000075021	.525	.0008714	6.10	11.63	11.64	.4801	1060.1	1.404
3	.04409	.02166	.000078851	.552	.0009179	6.43	11.65	11.67	.7201	1060.6	1.694
4	.04645	.02282	.000082890	.580	.0009671	6.77	11.68	11.70	.9601	1061.0	1.986
5	0.04886	0.02400	0.000087005	0.609	0.001017	7.12	11.70	11.72	1.200	1061.5	2.280
6	.05144	.02527	.000091399	.640	.001071	7.50	11.73	11.75	1.440	1061.9	2.577
7	.05412	.02658	.000095955	.672	.001127	7.89	11.75	11.77	1.680	1062.4	2.877
8	.05692	.02796	.00010070	.705	.001186	8.30	11.78	11.80	1.920	1062.8	3.180
9	.05988	.02941	.00010572	.740	.001247	8.73	11.80	11.83	2.160	1063.3	3.486
10	0.06295	0.03092	0.00011090	0.776	0.001311	9.18	11.83	11.85	2.400	1063.7	3.795
11	.06618	.03251	.00011634	.814	.001379	9.65	11.86	11.88	2.640	1064.2	4.108
12	.06958	.03418	.00012206	.854	.001450	10.15	11.88	11.91	2.880	1064.6	4.424
13	.07309	.03590	.00012794	.896	.001523	10.66	11.91	11.93	3.120	1065.1	4.742
14	.07677	.03771	.00013410	.939	.001600	11.20	11.93	11.96	3.359	1065.5	5.064
15	0.08067	0.03963	0.00014062	0.984	0.001682	11.77	11.96	11.99	3.599	1066.0	5.392
16	.08469	.04160	.00014732	1.031	.001766	12.36	11.98	12.01	3.839	1066.4	5.722
17	.08895	.04369	.00015440	1.081	.001855	12.99	12.00	12.04	4.079	1066.9	6.058
18	.09337	.04586	.00016174	1.132	.001947	13.63	12.03	12.07	4.319	1067.3	6.397
19	.09797	.04812	.00016935	1.185	.002043	14.30	12.06	12.09	4.559	1067.8	6.741
20	0.1028	0.05050	0.00017747	1.242	0.002144	15.01	12.08	12.12	4.798	1068.2	7.088
21	.1078	.05295	.00018564	1.299	.002250	15.75	12.11	12.15	5.038	1068.7	7.443
22	.1132	.05560	.00019439	1.361	.002361	16.53	12.13	12.18	5.278	1069.1	7.802
23	.1186	.05826	.00020335	1.423	.002476	17.33	12.16	12.20	5.518	1069.6	8.166
24	.1244	.06111	.00021276	1.489	.002596	18.17	12.18	12.23	5.758	1070.0	8.536
25	0.1304	0.06405	0.00022255	1.558	0.002722	19.05	12.21	12.26	5.998	1070.5	8.912
26	.1366	.06710	.00023278	1.629	.002853	19.97	12.23	12.29	6.237	1070.9	9.292
27	.1432	.07034	.00024342	1.704	.002991	20.94	12.26	12.32	6.477	1071.4	9.682
28	.1500	.07368	.00025445	1.781	.003133	21.93	12.28	12.34	6.717	1071.8	10.075
29	.1571	.07717	.00026597	1.862	.003283	22.99	12.31	12.37	6.957	1072.3	10.477
30	0.1645	0.08080	0.00027797	1.946	0.003439	24.07	12.33	12.40	7.197	1072.7	10.886
31	.1722	.08458	.00029043	2.033	.003601	25.21	12.36	12.43	7.437	1073.2	11.302
32	.1803	.08856	.00030343	2.124	.003771	26.40	12.38	12.46	7.677	1073.6	11.726
33	.1879	.09230	.00031471	2.203	.003931	27.52	12.41	12.49	7.917	1074.1	12.139
34	.1957	.09610	.00032690	2.288	.004094	28.66	12.43	12.51	8.157	1074.5	12.556

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp, F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds	Grains	Pounds	Grains					
35	0.20360	0.1000	0.0003394	2.376	0.004262	29.83	12.46	12.54	8.397	1075.0	12.979
36	.21195	.1041	.0003527	2.469	.004438	31.07	12.48	12.57	8.636	1075.4	13.409
37	.22050	.1083	.0003662	2.563	.004618	32.33	12.51	12.60	8.876	1075.9	13.845
38	.22925	.1126	.0003799	2.660	.004803	33.62	12.53	12.63	9.116	1076.3	14.285
39	.23842	.1171	.0003943	2.760	.004996	34.97	12.56	12.66	9.356	1076.8	14.736
40	0.24778	0.1217	0.0004090	2.863	0.005194	36.36	12.59	12.69	9.596	1077.2	15.191
41	.25755	.1265	.0004243	2.970	.005401	37.80	12.61	12.72	9.836	1077.7	15.657
42	.26773	.1315	.0004401	3.081	.005616	39.31	12.64	12.75	10.08	1078.1	16.13
43	.27832	.1367	.0004566	3.196	.005840	40.88	12.66	12.78	10.32	1078.6	16.62
44	.28911	.1420	.0004735	3.315	.006069	42.48	12.69	12.81	10.56	1079.0	17.11
45	0.30031	0.1475	0.0004909	3.436	0.006306	44.14	12.71	12.84	10.80	1079.5	17.61
46	.31191	.1532	.0005088	3.562	.006553	45.87	12.74	12.87	11.04	1079.9	18.12
47	.32393	.1591	.0005274	3.692	.006808	47.66	12.76	12.90	11.28	1080.4	18.64
48	.33635	.1652	.0005465	3.826	.007072	49.50	12.79	12.93	11.52	1080.8	19.16
49	.34917	.1715	.0005663	3.964	.007345	51.42	12.81	12.96	11.76	1081.3	19.70
50	0.36241	0.1780	0.0005866	4.106	0.007626	53.38	12.84	12.99	12.00	1081.7	20.25
51	.37625	.1848	.0006078	4.255	.007921	55.45	12.86	13.02	12.23	1082.2	20.80
52	.39051	.1918	.0006296	4.407	.008226	57.58	12.89	13.06	12.47	1082.6	21.38
53	.40496	.1989	.0006516	4.561	.008534	59.74	12.91	13.09	12.71	1083.1	21.95
54	.42003	.2063	.0006746	4.722	.008856	61.99	12.94	13.12	12.95	1083.5	22.55
55	0.43570	0.2140	0.0006984	4.889	0.009192	64.34	12.96	13.15	13.19	1084.0	23.15
56	.45179	.2219	.0007228	5.060	.009536	66.75	12.99	13.19	13.43	1084.4	23.77
57	.46828	.2300	.0007477	5.234	.009890	69.23	13.01	13.22	13.67	1084.9	24.40
58	.48538	.2384	.0007735	5.415	.01026	71.82	13.04	13.25	13.91	1085.3	25.05
59	.50310	.2471	.0008003	5.602	.01064	74.48	13.06	13.29	14.15	1085.8	25.70
60	0.52142	0.2561	0.0008278	5.795	0.01103	77.21	13.09	13.32	14.39	1086.2	26.37
61	.54035	.2654	.0008562	5.993	.01144	80.08	13.11	13.35	14.63	1086.7	27.06
62	.55970	.2749	.0008852	6.196	.01186	83.02	13.14	13.39	14.87	1087.1	27.76
63	.57985	.2848	.0009153	6.407	.01229	86.03	13.16	13.42	15.11	1087.6	28.48
64	.60042	.2949	.0009460	6.622	.01274	89.18	13.19	13.46	15.35	1088.0	29.21
65	0.62179	0.3054	0.0009778	6.845	0.01320	92.40	13.21	13.49	15.59	1088.5	29.96
66	.64378	.3162	.0010105	7.074	.01368	95.76	13.24	13.53	15.83	1088.9	30.73
67	.66638	.3273	.0010440	7.308	.01417	99.19	13.26	13.57	16.07	1089.4	31.51
68	.68980	.3388	.0010816	7.571	.01468	102.8	13.29	13.60	16.31	1089.8	32.31
69	.71382	.3506	.0011140	7.798	.01520	106.4	13.31	13.64	16.55	1090.3	33.12

[a] Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
			per Cu Ft		per lb of Dry Air						
	In. of Hg	Lb per Sq In.	Pounds	Grains	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
70	0.73866	0.3628	0.0011507	8.055	0.01574	110.2	13.34	13.68	16.79	1090.7	33.96
71	.76431	.3754	.0011884	8.319	.01631	111.2	13.37	13.71	17.03	1091.2	34.83
72	.79058	.3883	.0012269	8.588	.01688	118.2	13.40	13.75	17.27	1091.6	35.70
73	.81766	.4016	.0012667	8.867	.01748	122.4	13.42	13.79	17.51	1092.1	36.60
74	.84555	.4153	.0013075	9.153	.01809	126.6	13.44	13.83	17.75	1092.5	37.51
75	0.87448	0.4295	0.0013497	9.448	0.01873	131.1	13.47	13.87	17.99	1093.0	38.46
76	.90398	.4440	.0013927	9.749	.01938	135.7	13.49	13.91	18.23	1093.4	39.42
77	.93452	.4590	.0014371	10.06	.02005	140.4	13.52	13.95	18.47	1093.9	40.40
78	.96588	.4744	.0014825	10.38	.02075	145.3	13.54	13.99	18.71	1094.3	41.42
79	.99825	.4903	.0015295	10.71	.02147	150.3	13.57	14.03	18.95	1094.8	42.46
80	1.0316	0.5067	0.0015777	11.04	0.02221	155.5	13.59	14.08	19.19	1095.2	43.51
81	1.0661	.5236	.0016273	11.39	.02298	160.9	13.62	14.12	19.43	1095.7	44.61
82	1.1013	.5409	.0016781	11.75	.02377	166.4	13.64	14.16	19.67	1096.1	45.72
83	1.1377	.5588	.0017304	12.11	.02459	172.1	13.67	14.21	19.91	1096.6	46.88
84	1.1752	.5772	.0017841	12.49	.02543	178.0	13.69	14.26	20.15	1097.0	48.05
85	1.2135	0.5960	0.0018389	12.87	0.02629	184.0	13.72	14.30	20.39	1097.5	49.24
86	1.2527	.6153	.0018950	13.27	.02718	190.3	13.74	14.34	20.63	1097.9	50.47
87	1.2933	.6352	.0019531	13.67	.02810	196.7	13.77	14.39	20.87	1098.4	51.74
88	1.3346	.6555	.0020116	14.08	.02904	203.3	13.79	14.44	21.11	1098.8	53.02
89	1.3774	.6765	.0020725	14.51	.03002	210.1	13.82	14.48	21.35	1099.3	54.35
90	1.4211	0.6980	0.0021344	14.94	0.03102	217.1	13.84	14.53	21.59	1099.7	55.70
91	1.4661	.7201	.0021982	15.39	.03205	224.4	13.87	14.58	21.83	1100.2	57.09
92	1.5125	.7429	.0022634	15.84	.03312	231.8	13.89	14.63	22.07	1100.6	58.52
93	1.5600	.7662	.0023304	16.31	.03421	239.5	13.92	14.69	22.32	1101.1	59.99
94	1.6088	.7902	.0023992	16.79	.03535	247.5	13.94	14.73	22.56	1101.5	61.50
95	1.6591	0.8149	0.0024697	17.28	0.03652	255.6	13.97	14.79	22.80	1102.0	63.05
96	1.7108	.8403	.0025425	17.80	.03772	264.0	13.99	14.84	23.04	1102.4	64.62
97	1.7638	.8663	.0026164	18.31	.03896	272.7	14.02	14.90	23.28	1102.9	66.25
98	1.8181	.8930	.0026925	18.85	.04024	281.7	14.04	14.95	23.52	1103.3	67.92
99	1.8741	.9205	.0027700	19.39	.04156	290.9	14.07	15.01	23.76	1103.8	69.63
100	1.9316	0.9487	0.0028506	19.95	0.04293	300.5	14.10	15.07	24.00	1104.2	71.40
101	1.9904	.9776	.0029316	20.52	.04433	310.3	14.12	15.12	24.24	1104.7	73.21
102	2.0507	1.0072	.0030156	21.11	.04577	320.4	14.15	15.18	24.48	1105.1	75.06
103	2.1128	1.0377	.0031017	21.71	.04726	330.8	14.17	15.25	24.72	1105.6	76.97
104	2.1763	1.0689	.0031887	22.32	.04879	341.5	14.20	15.31	24.96	1106.0	78.92

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft Pounds	per Cu Ft Grains	per lb of Dry Air Pounds	per lb of Dry Air Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
105	2.2414	1.1009	0.0032786	22.95	0.05037	352.6	14.22	15.37	25.20	1106.5	80.93
106	2.3084	1.1338	.0033715	23.60	.05200	364.0	14.25	15.44	25.44	1106.9	83.00
107	2.3770	1.1675	.0034650	24.26	.05368	375.8	14.27	15.50	25.68	1107.4	85.13
108	2.4473	1.2020	.0035612	24.93	.05541	387.9	14.30	15.57	25.92	1107.8	87.30
109	2.5196	1.2375	.0036603	25.62	.05719	400.3	14.32	15.64	26.16	1108.3	89.54
110	2.5939	1.274	0.0037622	26.34	0.05904	413.3	14.35	15.71	26.40	1108.7	91.86
111	2.6692	1.311	.0038669	27.07	.06092	426.4	14.37	15.78	26.64	1109.2	94.21
112	2.7486	1.350	.0039729	27.81	.06292	440.4	14.39	15.85	26.88	1109.6	96.70
113	2.8280	1.389	.0040816	28.57	.06493	454.5	14.42	15.93	27.12	1110.1	99.20
114	2.9094	1.429	.0041911	29.34	.06700	469.0	14.45	16.00	27.36	1110.5	101.76
115	2.9929	1.470	0.0043047	30.13	0.06913	483.9	14.47	16.08	27.60	1111.0	104.40
116	3.0784	1.512	.0044208	30.95	.07134	499.4	14.50	16.16	27.84	1111.4	107.13
117	3.1660	1.555	.0045372	31.76	.07361	515.3	14.52	16.24	28.08	1111.9	109.92
118	3.2576	1.600	.0046620	32.63	.07600	532.0	14.55	16.32	28.32	1112.3	112.85
119	3.3492	1.645	.0047846	33.49	.07840	548.8	14.57	16.41	28.56	1112.8	115.80
120	3.4449	1.692	0.0049115	34.38	0.08093	566.5	14.60	16.50	28.80	1113.2	118.89
121	3.5406	1.739	.005040	35.28	.08348	584.4	14.62	16.58	29.04	1113.7	122.01
122	3.6404	1.788	.005173	36.21	.08616	603.1	14.65	16.68	29.28	1114.1	125.27
123	3.7422	1.838	.005311	37.18	.08892	622.4	14.67	16.77	29.52	1114.6	128.63
124	3.8460	1.889	.005450	38.15	.09175	642.3	14.70	16.87	29.76	1115.0	132.06
125	3.9519	1.941	0.005590	39.13	0.09466	662.6	14.72	16.96	30.00	1115.5	135.59
126	4.0618	1.995	.005734	40.14	.09770	683.9	14.75	17.06	30.24	1115.9	139.26
127	4.1718	2.049	.005882	41.17	.1008	705.6	14.77	17.17	30.48	1116.4	143.01
128	4.2858	2.105	.006031	42.22	.1040	728.0	14.80	17.27	30.72	1116.8	146.87
129	4.4039	2.163	.006188	43.32	.1074	751.8	14.83	17.38	30.96	1117.3	150.96
130	4.5220	2.221	0.006344	44.41	0.1107	774.9	14.85	17.49	31.20	1117.7	154.93
131	4.6441	2.281	.006504	45.53	.1143	800.1	14.88	17.61	31.45	1118.2	159.26
132	4.7703	2.343	.006671	46.70	.1180	826.0	14.90	17.73	31.69	1118.6	163.68
133	4.8986	2.406	.006839	47.87	.1218	852.6	14.93	17.85	31.93	1119.1	168.24
134	5.0289	2.470	.007010	49.07	.1257	879.9	14.95	17.97	32.17	1119.5	172.89
135	5.1633	2.536	0.007185	50.30	0.1297	907.9	14.98	18.10	32.41	1120.0	177.67
136	5.2997	2.603	.007364	51.55	.1339	937.3	15.00	18.23	32.65	1120.4	182.67
137	5.4402	2.672	.007547	52.83	.1382	967.4	15.03	18.36	32.89	1120.9	187.80
138	5.5827	2.742	.007732	54.12	.1427	998.9	15.05	18.50	33.13	1121.3	193.14
139	5.7293	2.814	.007923	55.46	.1473	1031.1	15.08	18.65	33.37	1121.8	198.61

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft Pounds	per Cu Ft Grains	per lb of Dry Air Pounds	per lb of Dry Air Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
140	5.8779	2.887	0.008116	56.81	0.1521	1064.7	15.10	18.79	33.61	1122.2	201.30
141	6.0306	2.962	.008313	58.19	.1570	1099.0	15.13	18.91	33.85	1122.7	210.11
142	6.1874	3.039	.008516	59.61	.1622	1135.4	15.15	19.10	34.09	1123.1	216.26
143	6.3482	3.118	.008724	61.07	.1675	1172.5	15.18	19.26	34.33	1123.6	222.53
144	6.5111	3.198	.008933	62.53	.1730	1211.0	15.20	19.43	34.57	1124.0	229.02
145	6.6781	3.280	0.009148	64.04	0.1787	1250.9	15.23	19.60	34.81	1124.5	235.76
146	6.8471	3.363	.009366	65.56	.1846	1292.2	15.25	19.78	35.05	1124.9	242.71
147	7.0222	3.449	.009590	67.13	.1908	1335.6	15.28	19.96	35.29	1125.4	250.02
148	7.1993	3.536	.009817	68.72	.1971	1379.7	15.30	20.15	35.53	1125.8	257.43
149	7.3805	3.625	.010040	70.28	.2037	1425.9	15.33	20.35	35.77	1126.3	265.20
150	7.5658	3.716	0.010284	71.99	0.2105	1473.5	15.35	20.55	36.02	1126.7	273.19
151	7.7551	3.809	.010526	73.68	.2176	1523.2	15.38	20.76	36.26	1127.2	281.51
152	7.9485	3.904	.010772	75.40	.2250	1575.0	15.40	20.97	36.50	1127.6	290.21
153	8.1460	4.001	.011022	77.15	.2327	1628.9	15.43	21.20	36.74	1128.1	299.25
154	8.3476	4.100	.011279	78.95	.2407	1684.9	15.45	21.43	36.98	1128.5	308.61
155	8.5532	4.201	0.011539	80.77	0.2490	1743.0	15.48	21.67	37.22	1129.0	318.34
156	8.7650	4.305	.011807	82.65	.2577	1803.9	15.50	21.93	37.46	1129.4	328.51
157	8.9788	4.410	.012077	84.54	.2667	1866.9	15.53	22.19	37.70	1129.9	339.04
158	9.1986	4.518	.012354	86.48	.2761	1932.7	15.56	22.46	37.94	1130.3	350.02
159	9.4206	4.627	.012634	88.44	.2858	2000.6	15.58	22.74	38.18	1130.8	361.36
160	9.6486	4.739	0.012919	90.43	0.2961	2072.7	15.61	23.03	38.43	1131.2	373.38
161	9.8807	4.853	.013211	92.48	.3067	2146.9	15.63	23.33	38.67	1131.7	385.76
162	10.119	4.970	.013509	94.56	.3179	2225.3	15.66	23.65	38.91	1132.1	398.80
163	10.361	5.089	.013812	96.68	.3295	2306.5	15.68	23.98	39.15	1132.5	412.34
164	10.608	5.210	.014120	98.84	.3416	2391.2	15.71	24.33	39.39	1133.0	426.42
165	10.860	5.334	0.014434	101.0	0.3544	2480.8	15.73	24.69	39.63	1133.5	441.34
166	11.117	5.460	.014753	103.3	.3677	2573.9	15.76	25.07	39.87	1133.9	456.81
167	11.379	5.589	.015080	105.6	.3817	2671.9	15.78	25.46	40.11	1134.4	473.11
168	11.646	5.720	.015410	107.9	.3964	2774.8	15.81	25.88	40.35	1134.8	490.18
169	11.919	5.854	.015750	110.3	.4118	2882.6	15.83	26.31	40.59	1135.3	508.11
170	12.196	5.990	0.016092	112.6	0.4280	2996.0	15.86	26.77	40.83	1135.7	526.91
171	12.480	6.130	.016444	115.1	.4451	3115.7	15.88	27.24	41.07	1136.2	546.79
172	12.770	6.272	.016801	117.6	.4631	3241.7	15.91	27.74	41.32	1136.6	567.68
173	13.065	6.417	.017164	120.1	.4821	3374.7	15.93	28.28	41.56	1137.1	589.76
174	13.366	6.565	.017534	122.7	.5022	3515.4	15.96	28.84	41.80	1137.5	613.05

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONCLUDED)

Temp F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft: Pounds	per Cu Ft: Grains	per lb of Dry Air: Pounds	per lb of Dry Air: Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
175	13.674	6.716	0.017914	125.4	0.5235	3664.5	15.98	29.43	42.04	1138.0	637.78
176	13.985	6.869	.018294	128.1	.5459	3821.3	16.01	30.05	42.28	1138.4	663.73
177	14.303	7.025	.018684	130.8	.5697	3987.9	16.03	30.71	42.52	1138.9	691.35
178	14.627	7.184	.019080	133.6	.5949	4164.3	16.06	31.41	42.76	1139.3	720.53
179	14.954	7.345	.019477	136.3	.6215	4350.5	16.08	32.15	43.00	1139.8	751.39
180	15.290	7.510	0.019888	139.2	0.6501	4550.7	16.11	32.94	43.24	1140.2	784.48
181	15.632	7.678	.020304	142.1	.6805	4763.5	16.13	33.78	43.49	1140.7	819.74
182	15.981	7.849	.020729	145.1	.7131	4991.7	16.16	34.68	43.73	1141.1	857.45
183	16.337	8.024	.021159	148.1	.7481	5236.7	16.18	35.65	43.97	1141.6	898.00
184	16.697	8.201	.021598	151.2	.7854	5497.8	16.21	36.67	44.21	1142.0	941.14
185	17.066	8.382	0.022045	154.3	0.8258	5780.6	16.23	37.78	44.45	1142.5	987.93
186	17.440	8.566	.022497	157.5	.8693	6085.1	16.26	38.98	44.69	1142.9	1038.21
187	17.821	8.753	.022956	160.7	.9162	6413.4	16.28	40.27	44.93	1143.4	1092.51
188	18.210	8.944	.023424	164.0	.9673	6771.1	16.31	41.67	45.18	1143.8	1151.58
189	18.605	9.138	.023900	167.3	1.0227	7158.9	16.34	43.04	45.42	1144.3	1216.04
190	19.008	9.336	0.024384	170.7	1.083	7581.0	16.36	44.85	45.66	1144.7	1285.37
191	19.419	9.538	.024881	174.2	1.150	8050.0	16.39	46.68	45.90	1145.2	1362.88
192	19.839	9.744	.025380	177.7	1.224	8568.0	16.41	48.70	46.14	1145.6	1448.35
193	20.266	9.954	.025893	181.3	1.306	9142.0	16.44	50.93	46.38	1146.1	1543.19
194	20.702	10.168	.026413	184.9	1.397	9779.0	16.46	53.42	46.62	1146.5	1648.28
195	21.144	10.385	0.026939	188.6	1.499	10493.0	16.49	56.20	46.86	1147.0	1766.21
196	21.592	10.605	.027472	192.3	1.613	11291.0	16.51	59.31	47.10	1147.4	1897.86
197	22.048	10.829	.028019	196.1	1.742	12194.0	16.54	62.85	47.34	1147.9	2046.98
198	22.512	11.057	.028571	200.0	1.890	13230.0	16.56	66.88	47.59	1148.3	2217.88
199	22.984	11.289	.029129	203.9	2.061	14427.0	16.59	71.54	47.83	1148.8	2415.51
200	23.465	11.525	0.029700	207.9	2.261	15827.0	16.61	76.99	48.07	1149.2	2646.41

[a]Compiled by W. M. Sawdon. vapor pressures converted from *International Critical Tables*.

$$h'_{fg}\,(W_{t'} - W) = c_{p_a}\,(t - t') + c_{p_s}W\,(t - t') \tag{9a}$$

and using $c_{p_a} = 0.24$ and $c_{p_s} = 0.45$

$$h'_{fg}\,(W_{t'} - W) = (0.24 + 0.45W)\,(t - t') \tag{9b}$$

where

h'_{fg} = latent heat of vaporization at t', Btu per pound.

$(W_{t'} - W)$ = increase in vapor associated with 1 lb of dry air when it is saturated adiabatically from an initial dry-bulb temperature, t, and an initial vapor content, W, pounds.

Knowing any two of the three primary variables, t, t', or W, the third may be found from this equation for any process of adiabatic saturation.

TOTAL HEAT AND HEAT CONTENT

The total heat of a mixture of dry air and water vapor was originally defined by W. H. Carrier as

$$\Sigma = c_{p_a}\,(t - 0) + W\,[h'_{fg} + c_{p_s}\,(t - t')] \tag{10}$$

where

Σ = total heat of the mixture, Btu per pound of dry air.
c_{p_a} = mean specific heat at constant pressure of dry air.
c_{p_s} = mean specific heat at constant pressure of water vapor.
t = dry-bulb temperature, degrees Fahrenheit.
t' = wet-bulb temperature, degrees Fahrenheit.
W = weight of water vapor mixed with each pound of dry air, pounds.
h'_{fg} = latent heat of vaporization at t', Btu per pound.

Since this definition holds for any mixture of dry air and water vapor, the total heat of a mixture with a relative humidity of 100 per cent and at a temperature equal to the wet-bulb temperature (t') is

$$\Sigma' = c_{p_a}\,(t' - 0) + W_{t'}\,h'_{fg} \tag{11}$$

By equating Equation 10 to Equation 11, the equation for the adiabatic saturation process, Equation 9a, follows. This demonstrates that the adiabatic saturation process at constant wet-bulb temperature is also a process of constant total heat. In short, the total heat of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. This fact furnishes a convenient means of finding the total heat of an air-vapor mixture in any state.

Enthalpy

This total heat of an air-vapor mixture is not exactly equal to the true heat content or enthalpy of the mixture since the heat content of the liquid is not included in Equation 10. With the meaning of heat content in agreement with present practise in other branches of thermodynamics, the true heat content of a mixture of dry air and water vapor (with 0 F as the datum for dry air, and the saturated liquid at 32 F as the datum for the water vapor) is

$$h = c_{p_a}\,(t - 0) + W\,h_s = 0.24\,(t - 0) + W\,h_s \tag{12}$$

where

h = the heat content of the mixture, Btu per pound of dry air.
t = the dry-bulb temperature, degrees Fahrenheit.
W = the weight of vapor per pound of dry air, pounds.
h_s = the heat content of the vapor in the mixture, Btu per pound.

The heat content of the water vapor in the mixture may be found in steam charts or tables when the dry-bulb temperature and the partial pressure of the vapor are known. Or, since the heat content of steam at low partial pressures, whether super-heated or saturated, depends only upon temperature, the following empirical equation, derived from Properties of Saturated Steam by J. H. Keenan, Table 8, may be used:

$$h_s = 1059.2 + 0.45\,t' \tag{13}$$

Substituting this value of h_s in Equation 12, the heat content of the mixture is

$$h = 0.24\,(t - 0) + W\,(1059.2 + 0.45\,t') \tag{14}$$

Example 5. Find the total heat of an air-vapor mixture having a dry-bulb temperature of 85 F and a wet-bulb temperature of 70 F.

Solution. From Table 6, for saturation at the wet-bulb temperature, $W_{t'} = 0.01574$, and from Equation 14,

$$h = 0.24\,(70-0) + 0.01574\,[1059.2 + (0.45 \times 70)] = 33.96 \text{ Btu per pound dry air.}$$

By considering the temperatures in Table 6 to be wet-bulb readings, the enthalpy of any air-vapor mixture may be obtained from the last column in the table.

An energy equation can be written that applies, in general, to various air conditioning processes, and this equation can be used to determine the quantity of heat transferred during such processes. In the most general form, this equation may be explained with the aid of Fig. 1 as follows.

The rectangle may represent any apparatus, *e.g.*, a drier, humidifier, dehumidifier, cooling tower, or the like, by proper choice of the direction of the arrows.

In general, a mixture of air and water vapor, such as atmospheric air, enters the apparatus at 1 and leaves at 3. Water is supplied at some temperature, t_2. For the flow of 1 lb of dry air (with accompanying vapor) through the apparatus. provided there is no appreciable change in the elevation or velocity of the fluids and no mechanical energy delivered to or by the apparatus,

$$h_1 + E_h + (W_3 - W_1)\,h_2 = h_3 + R_c$$

or

$$E_h - R_c = h_3 - h_1 - (W_3 - W_1)\,h_2 \tag{15}$$

where

E_h = the quantity of heat supplied per pound of dry air, Btu.
R_c = the quantity of heat lost externally by heat transfer from the apparatus. Btu per pound of dry air.
W_1 = the weight of water vapor entering, per pound of dry air.
W_3 = the weight of water vapor leaving, per pound of dry air.
h_2 = the heat content of the water supplied at t_2, Btu per pound.

$h_3 - h_1$ = the increase in the heat content of the air-water vapor mixture in passing through the apparatus, Btu per pound of dry air

$= 0.24\ (t_3 - t_1) + W_3\ (1059.2 + 0.45\ t_3) - W_1\ (1059.2 + 0.45 t_1)$

The net quantity of heat added to or removed from air-water vapor mixtures in air conditioning work is frequently *approximated* by taking the differences in total heat at exit and entrance.

For example, in Fig. 1, an *approximate* result is

$$E_h - R_c = \Sigma_3 - \Sigma_1 \qquad (16)$$

where

Σ_3 = the total heat of the air-vapor mixture at exit, Btu per pound of dry air.
Σ_1 = the total heat of the air-vapor mixture at entrance, Btu per pound of dry air.

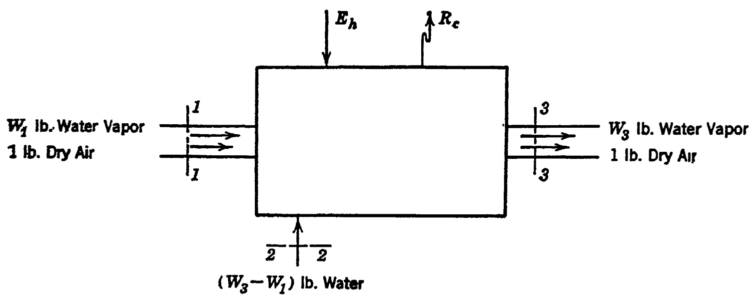

FIG. 1. DIAGRAM ILLUSTRATING ENERGY EQUATION 15

From the definitions of *total heat* and *heat content*, it may be demonstrated that Equation 16 is exactly equivalent to Equation 15, when, and only when, $t'_3 = t'_1 = t_2$; *i.e.*, when the initial and final wet-bulb temperatures and the temperature of the water supplied are equal. The one process that meets these conditions is adiabatic saturation, and either equation will give a result of zero; for other conditions, Equation 16 is approximate but satisfactory for many calculations.

The following problems illustrate the application of these principles:

Example 6. Heating (data from Example 2). Assuming the water to be supplied at 50 F, the net quantity of heat supplied is, from Equation 15,

$E_h - R_c = 0.24\ (70 - 0) + 0.000548 \times 0.45\ (70 - 0) + 0.005632$
$[1059.2 + 0.45 \times 70 - (50 - 32)] = 22.90$ Btu per pound of dry air.

Example 7. Cooling (data from Example 3). If the condensate is removed at 54 F the quantity of heat removed is found from Equation 15, by proper regard to the arrow direction in Fig. 1,

$E_h + R_c = 0.24\ (84 - 54) + 0.00887 \times 0.45\ (84 - 54) + 0.00361$
$[1059.2 + 0.45 \times 84 - (54 - 32)] = 11.24$ Btu per pound of dry air.

Using Table 6, the initial total heat of the air-vapor mixture, since the wet-bulb temperature is 70 F, is 33.96 Btu per pound of dry air.

The final total heat is, from Table 6, since the exit air is saturated, 22.55 Btu per pound. Hence, using Equation 16, the quantity of heat removed is, approximately, (33.96 − 22.55) or 11.41 Btu per pound of dry air. The degree of approximation to the correct result is evident in this example.

PSYCHROMETRIC CHART

The revised Bulkeley Psychrometric Chart[4], will be found attached to the inside back cover. It shows graphically the relationships expressed in Equations 9a and 9b. It also gives the grains of moisture per pound of dry air for saturation, the grains of moisture per cubic foot of saturated air, the total heat in Btu per pound of dry air saturated with moisture, and the weight of the dry air in pounds per cubic foot. Fig. 2 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

A is the total heat in Btu contained in the mixture above 0 F, and is to be referred to the column of figures at the left side of the chart. Heat of the liquid is not included.

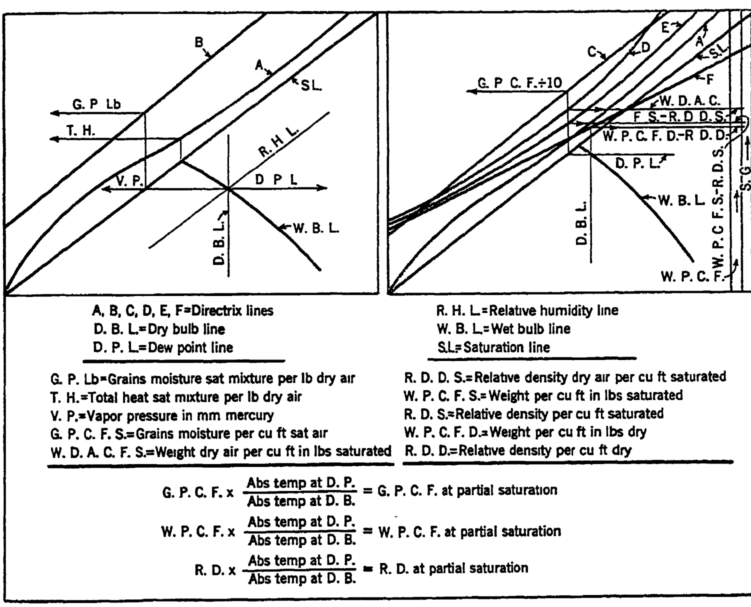

FIG. 2 DIAGRAMS SHOWING PROCEDURE TO FOLLOW IN USING BULKELEY CHART

B is the grains of moisture of water vapor contained in each pound of the saturated mixture and is to be referred to the figures at the left side of the chart.

C is the grains of moisture of water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart which are to be divided by 10.

D is the weight in decimal fractions of a pound of dry air in one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170 and 180 F. The relative density of the mixture is read in a similar manner from the same curve by the column of figures between the vertical dry-bulb temperature lines 180 and 190 F.

E is similar to *D* but is for one cubic foot of the saturated mixture.

[4]The Bulkeley Psychrometric Chart was presented to the Society in 1926. (See A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 163.) Single copy of the chart can be furnished at a cost of $ .25.

F is similar to *D* but is for dry air, devoid of all moisture or water vapor. For convenience, the approximate absolute temperature of 500 F is given at 40 F on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.

METHOD OF USING THE CHART

Example 8. Relative Humidity: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 46 per cent.

Example 9. Dew Point: At the intersection of the 78 F wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70.9 F.

Example 10. Vapor Pressure: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, pass in a horizontal direction to the left of the chart and on the logarithmic scale read the vapor pressure as 19.4 millimeters of mercury. (Divide by 25.4 for inches.)

Example 11. Total Heat Above 0 F in Mixture per Pound of Dry Air Saturated with Moisture: From where the wet-bulb line joins the saturation line, pass in a vertical direction on the 78 F dry-bulb line to its intersection with curve *A* and on the logarithmic scale at the left of the chart read 40.6 Btu per pound of mixture. The use of this curve to obtain the total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of Btu required to heat the mixture and humidify it, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

Example 12. Grains of Moisture per Pound of Dry Air Saturated with Moisture: From 70.9 F dew-point temperature on the saturation line, pass vertically to the intersection with curve *B* and on the logarithmic scale at the left read 114 grains of moisture per pound.

Example 13. Grains of Moisture per Cubic Foot of Mixture, Partially Saturated: From 70.9 F dew-point temperature on the saturation line proceed in a vertical direction to curve *C*, and on the logarithmic scale to the left read 83.3 which, divided by 10, gives 8.33 grains. A temperature of 70.9 F is equal to an absolute temperature of 530.9, and 95 F equals 555, absolute temperature. Therefore, $\frac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture.

Example 14. Grains of Moisture per Cubic Foot of Saturated Air: Starting at the saturation line at the desired temperature, pass in a vertical direction to curve *C* and on the logarithmic scale at the left, read a number which, divided by 10, will give the answer.

Example 15. Weight per Cubic Foot of Dry Air and Relative Density: From the point where, for example, the 70 F vertical dry-bulb line intersects curve *E*, pass to right side and read 0.075 lb; if cubic feet per pound are desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

Example 16. Weight of Dry Air per Cubic Foot of Saturated Mixture and Relative Density: From the point where, for example, the 70 F vertical line intersects the curve *D*, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70 F.

Example 17. Weight of Dry Air per Cubic Foot and Relative Density of Partially Saturated Air: Air at 50 F and a wet-bulb temperature of 46 F is to be heated to 130 F. The wet- and dry-bulb lines intersect at a dew-point temperature of 42 F. Pass to the left where this dew-point line intersects the saturation line and then pass in a vertical direction to where the 42 F dry-bulb line intersects with curve *D*. Then pass directly to the right and read the weight per cubic foot of saturated air at 42 F as 0.07844 and the relative density as 1.046. The absolute temperature at 42 F is 502, and at 130 F is 590. Therefore, $\frac{502}{590} = 0.851$. The weight of 1 cu ft of air at 50 F dry-bulb and 46 F wet-bulb when heated to 130 F is $0.07844 \times 0.851 = 0.06675$, and the relative density is $1.046 \times 0.851 = 0.89$.

PROPERTIES OF WATER

Composition of Water. Water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and 16 parts by weight of oxygen.

Density of Water. Water has its greatest density at 39.2 F, and it expands when heated or cooled from this temperature. At 62 F a U. S. gallon of 231 cu in. of water weighs approximately 8⅓ lb, and a cubic foot of water is equal to 7.48 gal. The specific volume of water depends on the temperature and it is always the reciprocal of its density. (See Table 7.)

TABLE 7. THERMAL PROPERTIES OF WATER

Temperature Deg F	Sat. Press. Lb per Sq In.	Volume Cu Ft per Lb	Weight Lb per Cu Ft	Specific Heat
32	0.0887	0.01602	62.42	1.0093
40	0.1217	0.01602	62.42	1.0048
50	0.1780	0.01602	62.42	1.0015
60	0.2561	0.01603	62.38	0.9995
70	0.3628	0.01605	62.31	0.9982
80	0.5067	0.01607	62.23	0.9975
90	0.6980	0.01610	62.11	0.9971
100	0.9487	0.01613	62.00	0.9970
110	1.274	0.01616	61.88	0.9971
120	1.692	0.01620	61.73	0.9974
130	2.221	0.01625	61.54	0.9978
140	2.887	0.01629	61.39	0.9984
150	3.716	0.01634	61.20	0.9990
160	4.739	0.01639	61.01	0.9998
170	5.990	0.01645	60.79	1.0007
180	7.510	0.01650	60.61	1.0017
190	9.336	0.01656	60.39	1.0028
200	11.525	0.01663	60.13	1.0039
210	14.123	0.01669	59.92	1.0052
212	14.696	0.01670	59.88	1.0055
220	17.188	0.01676	59.66	1.0068
240	24.97	0.01690	59.17	1.0104
260	35.43	0.01706	58.62	1.0148
280	49.20	0.01723	58.04	1.0200
300	67.01	0.01742	57.41	1.0260
350	134.62	0.01797	55.65	1.0440
400	247.25	0.01865	53.62	1.0670
450	422.61	0.01950	51.30	1.0950
500	681.09	0.02050	48.80	1.1300
550	1045.4	0.02190	45.70	1.2000
600	1544.6	0.02410	41.50	1.3620
700	3096.4	0.03940	25.40	....

Water Pressures. Pressures are often stated in feet or inches of water column. At 62 F, with h equal to the head in feet, the pressure of a column of water is 62.383h lb per square foot, or 0.433h lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of one pound per square inch at 62 F.

Boiling Point of Water. The boiling point of water varies with the pressure; it is lower at higher altitudes. A change in pressure will always be accompanied by a change in the boiling point, and there will be a cor-

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume: Sat. Liquid	Specific Volume: Evap.	Specific Volume: Sat. Vapor	Total Heat: Sat. Liquid	Total Heat: Evap.	Total Heat: Sat. Vapor	Entropy: Sat. Liquid	Entropy: Evap.	Entropy: Sat. Vapor	Abs. Press. Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
½" Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	½" Hg
¾" Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	¾" Hg
1" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1" Hg
1½" Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	1½" Hg
2" Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	2" Hg
2½" Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	2½" Hg
3" Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3" Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	96.0
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0

[a]Abstracted from *Steam Tables and Mollier Diagram*, by Prof. J. H. Keenan, 1930 edition, by permission the publisher, *The American Society of Mechanical Engineers*.

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume Sat. Liquid	Specific Volume Evap.	Specific Volume Sat. Vapor	Total Heat Sat. Liquid	Total Heat Evap.	Total Heat Sat. Vapor	Entropy Sat. Liquid	Entropy Evap.	Entropy Sat. Vapor	Abs. Press. Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Concluded)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume Sat. Liquid	Evap.	Sat. Vapor	Total Heat Sat. Liquid	Evap.	Sat. Vapor	Entropy Sat. Liquid	Evap.	Sat. Vapor	Abs. Press. Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	250.0
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	260.0
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	270.0
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	280.0
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	290.0
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	300.0
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	320.0
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	340.0
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	360.0
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	380.0
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400.0
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	420.0
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	440.0
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	460.0
480.0	462.80	0.0197	0.9456	0.9633	444.9	759.0	1203.9	0.6441	0.8228	1.4670	480.0
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500.0
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	520.0
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	540.0
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	560.0
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	580.0
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	600.0
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	620.0
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	640.0
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	660.0
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	680.0
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	700.0
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	720.0
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	740.0
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	760.0
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	780.0
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	800.0
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	820.0
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	840.0
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	860.0
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	880.0
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900.0
920.0	534.56	0.0214	0.4635	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	920.0
940.0	537.13	0.0215	0.4520	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	940.0
960.0	539.66	0.0216	0.4409	0.4625	539.6	651.5	1191.1	0.7410	0.6520	1.3930	960.0
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	980.0
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000.0
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050.0
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100.0
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150.0
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200.0
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250.0
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300.0
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350.0
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400.0
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450.0
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500.0
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600.0
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700.0
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800.0
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	1900.0
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	2000.0
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200.0
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	2400.0
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600.0
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	2800.0
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000.0
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	3200.0
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3226.0

responding change in the latent heat of evaporation. These values are given in Table 8.

Specific Heat. The specific heat of water, or the amount of heat (Btu) required to raise the temperature of one pound of water one degree Fahrenheit, varies with the temperature, but it is commonly assumed to be unity at all temperatures. Steam tables are based on exact values, however. The specific heat of ice at 32 F is 0.492 Btu per pound. The amount of heat required to raise one pound of water at 32 F through a known temperature interval depends on the average specific heat for the temperature range.

Sensible and Latent Heat. The heat necessary to raise the temperature of one pound of water from 32 F to the boiling point is known as the *heat of the liquid* or *sensible heat.* When more heat is added, the water begins to evaporate and expand at constant temperature until the water is entirely changed into steam. The heat thus added is known as the *latent heat of evaporation.*

PROPERTIES OF STEAM

Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. This change in state takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The volume of a pound of steam is the *specific volume* which decreases as the pressure increases. The reciprocal of this, or the weight of steam per cubic foot, is the *density.* (See Table 8.)

Steam which is in contact with the water from which it was generated is known as *saturated steam.* If it contains no actual water in the form of mist or priming, it is called *dry saturated steam.* If this be heated and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *super-heated,* that is, its temperature will be higher than that of saturated steam at the same pressure.

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PROBLEMS IN PRACTICE

1 ● Given air at 70 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.00 in. Hg, find the weight of vapor per pound of dry air.

Pressure of saturated vapor = e_t = 0.7387 in. Hg (Table 6).
From Equation 5a,

$$W = 0.622 \left(\frac{0.5 \times 0.7387}{29.00 - (0.5) \times (0.7387)}\right)$$

W = 0.008024 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

Approximate Method:

Weight of saturated vapor per pound of dry air = W_t = 0.01574 lb (Table 6). 0.01574 $\times$ 0.5 = 0.00787 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

2 ● Given air with a dry-bulb temperature of 80 F, relative humidity of 55 per cent, and a barometric pressure of 28.85 in. Hg, calculate the weight of a cubic foot of mixture.

Pressure of saturated vapor at 80 F = e_t = 1.0316 in. Hg (Table 6).

Pressure of the vapor in the mixture = 1.0316 $\times$ 0.55 = 0.5676 in. Hg.

Pressure of the dry air in the mixture = 28.85 $-$ 0.5676 = 28.282 in. Hg.

$pV = wR\,(t + 460)$ (R = 0.753 when partial pressure of air is expressed in in. Hg).
$28.282 \times 1 = d_a \times 0.753 \times (80 + 460)$

$$d_a = \frac{28.282}{0.753 \times 540} = 0.06955 \text{ lb}$$ = weight of dry air in 1 cu ft of the mixture.

Likewise from Equation 4a,

$$d_v = \frac{0.5676}{1.21 \times 540} = 0.000868 \text{ lb}$$ = weight of vapor per cubic foot at 55 per cent relative humidity.

Weight of 1 cu ft of the mixture = 0.06955 + 0.000868 = 0.070418 lb.

3 ● Given air with a dry-bulb temperature of 75 F, a relative humidity of 60 per cent, and a barometric pressure of 28.80 in. Hg, calculate the volume of 1 lb of the mixture.

Pressure of saturated vapor at 75 F = e_t = 0.8745 in. Hg.

Pressure of vapor in the mixture = 0.8745 $\times$ 0.6 = 0.525 in. Hg.

Pressure of dry air in the mixture = 28.80 $-$ 0.525 = 28.275 in. Hg.

$$d_a = \frac{28.275}{0.753 \times 535} = 0.07018 \text{ lb} = \text{weight of dry air in 1 cu ft of the mixture.}$$

From Equation 4a,

$$d_v = \frac{0.525}{1.21 \times 535} = 0.000811 \text{ lb} = \text{weight of vapor per cubic foot at 55 per cent}$$

relative humidity.

Weight of 1 cu ft of the mixture = 0.07018 + 0.000811 = 0.070991 lb.

Volume of 1 lb of the mixture $= \frac{1}{0.070991} = 14.08$ cu ft.

4 ● It is desired to maintain a temperature of 80 F and a relative humidity of 50 per cent in a factory where the equipment gives off 6,000 Btu per hour. If the entering air is at 70 F with an average barometric pressure of 29.92 in. Hg; determine the relative humidity, and the pounds of air required per hour if there is no heat interchange between the walls, windows, or floors of the building.

Pressure of saturated vapor at 80 F = 1.0316 in. Hg (Table 6).

Pressure of vapor in the mixture = 1.0316 × 0.5 = 0.5158 in. Hg.

$$W = 0.622 \left(\frac{0.5158}{29.92 - 0.5158}\right) = 0.01091 \text{ lb.}$$

Pressure of saturated vapor at 70 F = 0.7387 in. Hg.

With the same specific humidity

$$0.01091 = 0.622 \left(\frac{0.7387 \times \Phi}{29.92 - (0.7387 \times \Phi)}\right)$$

Φ = 69.8 per cent relative humidity at 70 F.

$h = 0.24 \times 80 + 0.01091\,[1059.2 + (0.45 \times 80)] = 31.15$ Btu per pound, the heat content of the mixture at 80 F and 50 per cent relative humidity.

$h = 0.24 \times 70 + 0.01091\,[1059.2 + (0.45 \times 70)] = 28.70$ Btu per pound, the heat content of the mixture at 70 F and the same specific humidity.

31.15 − 28.70 = 2.45 Btu to be removed per pound of air.

6000 Btu = heat given off by equipment per hour.

$\frac{6000}{2.45} = 2449$ lb of air required per hour.

5 ● Given 1 lb of dry air at 78 F and a barometric pressure of 29.92 in. Hg; calculate the volume. If the temperature is raised to 96 F and the volume remains constant, what will be the new pressure, P_2, in in. Hg?

$PV = wR\,(t + 460)$

R (for air) = 53.34.

W = 1 lb.

P = absolute pressure, pounds per square foot.

$$V = \frac{1 \times 53.34 \times (78 + 460)}{29.92 \times 0.491 \times 144}$$

V = 13.57 cu ft = volume of 1 lb.

$$\frac{P_2}{P_1} = \frac{T_2}{T_1}; \quad P_2 = \frac{T_2 P_1}{T_1}$$

$$P_2 = \frac{(96 + 460)\,(29.92 \times 0.491 \times 144)}{(78 + 460)\,(0.491 \times 144)} = 30.90 \text{ in. Hg.}$$

6 ● Given saturated air at a temperature of 75 F and a barometric pressure of 29.92 in. Hg; determine the heat content of the mixture per pound of dry air, including the heat content of the liquid above 32 F.

From Equation 12,

$h = 0.24\ (t - 0) + Wh_s.$

where

$h_s = 1059.2 + 0.45t'$ (Empirical equation derived from Keenan's Steam Tables).

$t' = 75$ F.

$W = 0.01873$ lb of water vapor (Table 6).

$h = 0.24\ (75 - 0) + 0.01873\ [1059.2 + (0.45 \times 75)].$

$h = 38.46$ Btu per pound of dry air.

7 ● A building requires 50,000 cu ft of air per hour to be raised from −10 F dry-bulb and 75 per cent relative humidity to 72 F dry-bulb and 30 per cent relative humidity. Determine the amount of heat and the weight of water which it is necessary to supply per hour if the temperature of the supply water is 50 F and the barometric pressure is 28.75 in. Hg.

Assume air volume to be dry air at 70 F.

Weight of air = 0.075 × 50,000 = 3750 lb per hour.

From Table 6,

Pressure of vapor in the mixture, outside air = 0.75 × 0.0221 = 0.0166 in. Hg.

Specific humidity, outside air $= 0.622 \left(\dfrac{0.0166}{28.75 - 0.0166}\right) = 0.0003589$ lb.

From Table 6,

Pressure of vapor in the mixture, inside air = 0.30 × 0.7906 = 0.2372 in. Hg.

Specific humidity, inside air $= 0.622 \left(\dfrac{0.2372}{28.75 - 0.23718}\right) = 0.005174$ lb.

Water to be added = 3750 (0.005174 − 0.0003589) = 18.06 lb per hour.

Heat content, inside air = 0.24 × 72 + 0.005174 [1059.2 + (0.45 × 72)] = 22.925 Btu per pound.

Heat content, outside air = 0.24 × (−10) + 0.0003589 [1059.2 + (0.45 × −10)] = −2.021 Btu per pound.

Btu added incident to the water per pound of dry air.

(0.005174 − 0.0003589) (50 − 32) = 0.0867 Btu per pound.

Heat requirement per hour = [22.925 − (−2.021 + 0.0867)] × 3750 = 93,221 Btu.

8 ● Determine the amount of heat and water that must be extracted to cool 3750 lb of air (weighed dry) from 95 F and 60 per cent relative humidity to 50 F and 100 per cent relative humidity with a barometric pressure of 28.75 in. Hg.

Pressure of vapor in the mixture, outside air = 0.6 × 1.659 = 0.995 in. Hg.

Specific humidity, outside air $= 0.622 \left(\dfrac{0.995}{28.75 - 0.995}\right) = 0.0223$ lb.

Specific humidity, inside air = 0.007626 lb.

Weight of water to be extracted per hour = (0.0223 − 0.007626) × 3750 = 55.03 lb.

Heat content, outside air = 0.24 × 95 + 0.0223 [1059.2 + (0.45 × 95)] = 47.37 Btu per pound.

Heat content, inside air = 0.24 × 50 + 0.00764 [1059.2 + (0.45 × 50)] = 20.26 Btu per pound.

Heat to be extracted = (47.37 − 20.26) × 3750 = 101,662 Btu.

Chapter 2

REFRIGERANTS AND AIR DRYING AGENTS

Properties of Refrigerant Substances, Selection Factors, Solid Adsorbents, Liquid Absorbents, Nature of Processes, Temperature Pressure Concentration Relations

BOTH cooling and dehumidification of air are usually desirable at certain times. Cooling may be regarded as the extraction of sensible heat while dehumidification necessitates the removal of latent heat. By a suitable selection and combination of methods and equipment these two processes may be accomplished simultaneously or they may be secured independent of each other and at different times as desired. On occasion, the desired result can be secured by extracting sensible heat only while in other cases drying alone will produce the end conditions sought. Suitable substances must be available for use in every particular case.

Generally, when both cooling and dehumidification are sought current practice makes use of artificially produced refrigeration in some form. The substances used as the heat-carrying agents in these applications are called refrigerants. Air drying agents are substances which permit dehumidification of air as a process separate and independent of the extraction of sensible heat from it. Refrigerants and air drying agents are treated separately in this chapter where the aim is to present a statement of the properties of these substances which are of especial importance in air conditioning work. Little mention is made here of methods, systems, or equipment whereby these substances may be applied, which information may be found in Chapter 24.

REFRIGERANTS

Since air cooling and dehumidification are frequently accomplished by evaporating a liquid under circumstances which will permit the heat necessary to be extracted from the air, the refrigerants are substances which are capable of being changed into liquids or vapors within workable temperature and pressure ranges. There are many substances which might be so used but in practice the choice is limited by a wide variety of considerations including availability, cost, safety, chemical stability and adaptability to the type of refrigerating system to be used.

In this chapter detailed consideration is limited to six substances, viz: ammonia, carbon dioxide, dichlorodifluoromethane (F_{12}), methyl

TABLE 1. PROPERTIES OF AMMONIA

Sat. Temp. F	Abs Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746.5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631.8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1.464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693

TABLE 1. PROPERTIES OF AMMONIA—(Continued)

SAT. TEMP. F	ABS PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1.3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634 0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

chloride, water, and monofluorotrichloromethane (F_{11}), for each of which a table is presented. Each table gives the principal physical properties of the saturated substance, and all are arranged in uniform fashion. In each case columns are included which give the heat content and entropy of the superheated vapor at two selected points. Tables 1, 2, 3 and 4 which include the refrigerants much used in reciprocating and rotary mechanical compression systems have a 2 F temperature interval. As water and F_{11} are much used in centrifugal compression systems the temperature interval in Tables 5 and 6 is 5 F.

AIR DRYING AGENTS

Moisture may be removed from air, thus accomplishing dehumidification, by the use of any one of a number of substances if the moist air and these substances are brought together under suitable circumstances. One class of these substances are solids at ordinary conditions and have the power of adsorbing the moisture from the air. Another class of air drying agents are liquids under ordinary conditions and absorb the moisture from the air. Nearly all the drying agents in frequent commercial use in air conditioning installations are of one or the other of these two classes.

Adsorbents

These substances are characterized by a physical structure containing a great number of extremely small pores but still retaining sufficient mechanical strength to resist whatever wear and handling to which they are subjected. To be suitable for air drying purposes they must be widely available at economical cost, durable in use, stable in form and properties, and capable of withstanding the re-activation processes by which they are made ready for repeated use. They must also possess capacity for adsorbing and holding so sufficient a quantity of moisture that the dimensions of the beds necessary to accommodate them will be practical.

TABLE 2. PROPERTIES OF CARBON DIOXIDE

Sat. Temp. F	Abs Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From —40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	305.5	0.01570	0.29040	18.8	138.9	0.0418	0.3024	153.7	0.3342	167.5	0.3612
2	315.9	0.01579	0.28030	19.8	138.8	0.0440	0.3014	153.7	0.3330	167.6	0.3600
4	326.5	0.01588	0.27070	20.8	138.8	0.0461	0.3005	153.7	0.3318	167.7	0.3588
5	332.0	0.01592	0.26610	21.3	138.8	0.0472	0.3000	153.7	0.3312	167.7	0.3582
6	337.4	0.01596	0.26140	21.8	138.7	0.0483	0.2994	153.7	0.3306	167.8	0.3576
8	348.7	0.01605	0.25260	22.9	138.7	0.0504	0.2982	153.7	0.3293	167.9	0.3563
10	360.2	0.01614	0.24370	24.0	138.7	0.0526	0.2970	153.7	0.3281	168.0	0.3550
12	371.9	0.01623	0.23540	25.0	138.6	0.0548	0.2958	153.7	0.3270	168.1	0.3538
14	383.9	0.01632	0.22740	26.1	138.6	0.0571	0.2946	153.7	0.3259	168.2	0.3526
16	396.2	0.01642	0.21970	27.2	138.5	0.0593	0.2933	153.7	0.3249	168.3	0.3513
18	408.9	0.01652	0.21210	28.3	138.4	0.0616	0.2921	153.7	0.3238	168.5	0.3501
20	421.8	0.01663	0.20490	29.4	138.3	0.0638	0.2909	153.7	0.3227	168.6	0.3489
22	434.0	0.01673	0.19790	30.5	138.2	0.0662	0.2897	153.7	0.3214	168.7	0.3479
24	448.4	0.01684	0.19120	31.7	138.1	0.0686	0.2885	153.7	0.3202	168.8	0.3470
26	462.2	0.01695	0.18460	32.9	138.0	0.0710	0.2873	153.7	0.3189	168.9	0.3460
28	476.3	0.01707	0.17830	34.1	137.9	0.0734	0.2861	153.7	0.3177	169.0	0.3451
30	490.8	0.01719	0.17220	35.4	137.8	0.0758	0.2849	153.7	0.3164	169.1	0.3441
32	505.5	0.01731	0.16630	36.7	137.7	0.0781	0.2834	153.7	0.3158	169.2	0.3431
34	522.6	0.01744	0.16030	37.9	137.4	0.0804	0.2820	153.7	0.3151	169.3	0.3421
36	536.0	0.01759	0.15500	39.1	137.2	0.0828	0.2805	153.7	0.3145	169.4	0.3411
38	551.7	0.01773	0.14960	40.4	136.9	0.0851	0.2791	153.7	0.3138	169.5	0.3401
39	559.7	0.01780	0.14700	41.0	136.8	0.0862	0.2783	153.7	0.3135	169.5	0.3396
40	567.8	0.01787	0.14440	41.7	136.7	0.0874	0.2776	153.7	0.3132	169.6	0.3391
41	576.0	0.01794	0.14185	42.3	136.5	0.0887	0.2768	153.7	0.3127	169.6	0.3386
42	584.3	0.01801	0.13930	42.9	136.3	0.0899	0.2761	153.7	0.3122	169.7	0.3381
44	601.1	0.01817	0.13440	44.3	136.1	0.0924	0.2745	153.7	0.3112	169.8	0.3371
46	618.2	0.01834	0.12970	45.6	135.7	0.0950	0.2730	153.7	0.3101	169.9	0.3362
48	635.7	0.01851	0.12500	47.0	135.4	0.0975	0.2714	153.7	0.3091	170.0	0.3352
50	653.6	0.01868	0.12050	48.4	135.0	0.1000	0.2699	153.7	0.3081	170.1	0.3342
52	671.9	0.01887	0.11610	49.8	134.5	0.1027	0.2681	153.7	0.3069	170.2	0.3333
54	690.6	0.01906	0.11170	51.2	133.9	0.1054	0.2663	153.7	0.3057	170.3	0.3324
56	709.5	0.01927	0.10750	52.6	133.4	0.1081	0.2644	153.7	0.3046	170.5	0.3315
58	728.8	0.01948	0.10340	54.0	132.7	0.1108	0.2626	153.7	0.3034	170.6	0.3306
60	748.6	0.01970	0.09940	55.5	132.1	0.1135	0.2608	153.7	0.3022	170.7	0.3297
62	769.0	0.01995	0.09545	57.0	131.3	0.1164	0.2584	153.7	0.3012	170.8	0.3289
64	789.4	0.02020	0.09180	58.6	130.6	0.1194	0.2560	153.7	0.3002	170.9	0.3281
66	810.3	0.02048	0.08800	60.2	129.7	0.1223	0.2535	153.7	0.2991	171.0	0.3273
68	831.6	0.02079	0.08422	61.9	128.7	0.1253	0.2511	153.7	0.2981	171.1	0.3265
70	853.4	0.02112	0.08040	63.7	127.5	0.1282	0.2487	153.7	0.2971	171.2	0.3257
72	875.8	0.02152	0.07654	65.5	126.0	0.1321	0.2450	153.7	0.2962	171.3	0.3250
74	898.2	0.02192	0.07269	67.3	124.5	0.1360	0.2414	153.7	0.2953	171.4	0.3242
76	921.3	0.02242	0.06875	69.4	122.8	0.1398	0.2377	153.7	0.2945	171.5	0.3235
78	944.8	0.02300	0.06473	71.6	120.9	0.1437	0.2341	153.7	0.2936	171.6	0.3227
80	968.7	0.02370	0.06064	73.9	118.7	0.1476	0.2304	153.7	0.2927	171.7	0.3220
82	993.0	0.02456	0.05648	76.4	116.6	0.1578	0.2195	153.7	0.2920	173.8	0.3215
84	1017.7	0.02553	0.05223	79.4	113.9	0.1679	0.2087	153.7	0.2914	176.0	0.3209
86	1043.0	0.02686	0.04789	83.3	110.4	0.1781	0.1978	153.7	0.2907	178.2	0.3204
87.8	1069.9	0.03454	0.03454	97.0	97.0	0.1880	0.1880	153.7	0.2901	180.1	0.3199

Aluminum Oxide, (Alumina), in a porous, amorphous form is a solid adsorbent frequently called by the common name *activated alumina*, and containing small amounts of hydrated aluminum oxide, and very small amounts of soda, and various metallic oxides. A good grade of activated alumina will show 92 per cent of Al_2O_3, and its soda content will be combined with silica and alumina into an insoluble compound. This substance also has the property of adsorbing certain gases and certain vapors other

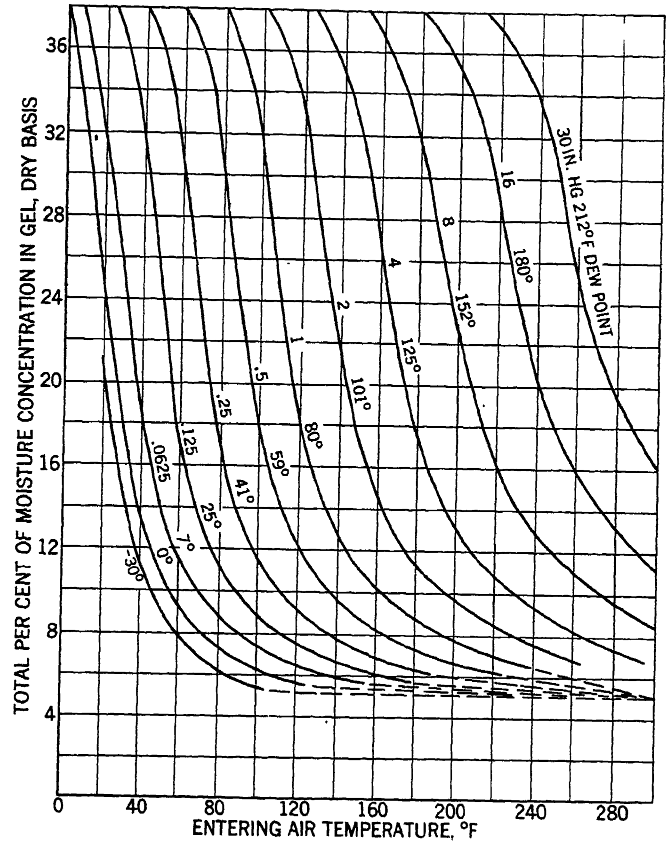

FIG. 1. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION RELATION FOR A SILICA GEL BED AT CONSTANT TEMPERATURE

than water vapor—a property which is sometimes useful in air conditioning installations. It is available commercially in granules ranging from a fine powder to pieces approximately 1.5 in. in diameter. It has high adsorptive capacity per unit of weight, and is non-toxic. It may be repeatedly re-activated after becoming saturated with adsorbed moisture without practical loss of its adsorptive ability. In the grade frequently used for air drying the re-activation may be accomplished at temperatures under 350 F. Specific gravity is 3.25 and the pores are reported to occupy 58 per cent of the volume of each particle. For most estimating purposes

TABLE 3. PROPERTIES OF DICHLORODIFLUOROMETHANE(F_{12})

Sat. Temp. F	Abs Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F: Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
2	24.89	0.0110	1.574	8.67	78.44	0.01961	0.17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0.02052	0.17060	82.17	0.17795	85.76	0.18511
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86.26	0.18477
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10.82	79.59	0.02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33.08	0.0112	1.207	11.70	80.05	0.02601	0.16974	83.61	0.17706	87.26	0.18413
18	34.40	0.0113	1.163	12.12	80.27	0.02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0.18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0.0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0.0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0.18291
36	48.13	0.0116	0.848	16.10	82.27	0.03502	0.16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0.819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0.0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90.28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0.0117	0.742	17.91	83.15	0.03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0.03948	0.16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04215	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0.0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92.28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0.04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0.16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.17425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
72	87.50	0.0121	0.479	24.37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106
74	90.20	0.0122	0.464	24.84	86.22	0.05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0.18091
78	95.85	0.0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94.57	0.18083
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0.378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0.0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0.17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998

TABLE 3. PROPERTIES OF DICHLORODIFLUOROMETHANE (F_{12})—Continued

SAT. TEMP. F	ABS PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
104	139.0	0.0128	0.302	32.15	88.95	0.06490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97.73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98.11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0 254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939
122	176.2	0.0132	0.233	36.66	90.28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0.227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0.214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	95.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204.8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

the volume-weight relation on a dry basis may be taken as 50 lb per cubic foot although in the smaller sizes the packed weight may be as much as 64 lb per cubic foot.

Silicon Dioxide, (Silica), in a special form obtained by suitably mixing sulphuric acid with sodium silicate, is another solid adsorbent and is commonly called *silica gel.* Its capillary structure is exceedingly small, so small that its exact structure has to be deduced rather than observed. The gel is available commercially in a wide variety of sizes of granules ranging from 4 to 300 mesh. It has high adsorptive capacity per unit of weight and is non-toxic, may be repeatedly re-activated without practical deterioration. Re-activation may be accomplished at temperatures of air up to 600 F although it is frequently accomplished with air or other gases at temperatures not over 350 F. Volume of the capillary pores is reported to be from 50 to 70 per cent of the total solid volume. For most estimating purposes the volume-weight relation can be assumed as from 38 to 40 lb per cubic foot on a dry basis.

Other substances having properties which make them available as solid adsorbents include lamisilate and charcoal but details of their physical properties are not available.

Nature of Adsorption Process

Adsorption is accomplished without chemical change between the air and the adsorbent substance. The adsorbent does not go into solution but water vapor is extracted from the air-vapor stream passing through the bed of adsorbent material and is caught and retained in the capillary pores. The exact nature of the process which goes on during adsorption

TABLE 4. PROPERTIES OF METHYL CHLORIDE

Sat. Temp. F	Abs Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	18.73	0.0162	5.052	14.4	192.4	0.0328	0.4197	215.6	0.467	237.2	0.507
2	19.60	0.0162	4.856	15.1	193.1	0.0344	0.4196	216.2	0.466	237.7	0.505
4	20.47	0.0163	4.661	15.8	193.8	0.0360	0.4195	216.7	0.465	238.2	0.504
5	20.91	0.0163	4.563	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
6	21.39	0.0163	4.476	16.6	194.4	0.0376	0.4194	217.3	0.464	238.8	0.502
8	22.34	0.0164	4.303	17.3	195.1	0.0391	0.4193	217.9	0.463	239.4	0.501
10	23.30	0.0164	4.129	18.1	195.8	0.0407	0.4192	218.5	0.463	240.0	0.500
12	24.38	0.0164	3.984	18.8	196.3	0.0423	0.4184	219.0	0.462	240.5	0.499
14	25.46	0.0164	3.839	19.6	196.7	0.0439	0.4176	219.5	0.462	241.0	0.498
16	26.55	0.0165	3.693	20.3	197.2	0.0454	0.4168	220.0	0.461	241.5	0.498
18	27.63	0.0165	3.548	21.1	197.6	0.0472	0.4160	220.5	0.461	242.0	0.497
20	28.71	0.0166	3.403	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0.496
22	29.98	0.0166	3.288	22.5	198.5	0.0501	0.4148	221.5	0.459	243.0	0.495
24	31.25	0.0166	3.172	23.3	198.9	0.0516	0.4143	222.0	0.459	243.6	0.495
26	32.53	0.0167	3.057	24.0	199.3	0.0532	0.4139	222.4	0.458	244.1	0.494
28	33.80	0.0167	2.941	24.8	199.7	0.0547	0.4134	222.9	0.458	244.7	0.494
30	35.07	0.0168	2.826	25.5	200.1	0.0562	0.4130	223.4	0.457	245.2	0.493
32	36.55	0.0168	2.734	26.2	200.5	0.0577	0.4124	223.9	0.456	245.7	0.492
34	38.03	0.0169	2.642	27.0	200.9	0.0592	0.4118	224.3	0.455	246.2	0.492
36	39.51	0.0169	2.549	27.7	201.4	0.0607	0.4111	224.8	0.455	246.7	0.491
38	40.99	0.0169	2.457	28.5	201.8	0.0622	0.4105	225.2	0.454	247.2	0.491
39	41.73	0.0170	2.411	28.8	202.0	0.0629	0.4102	225.5	0.453	247.4	0.490
40	42.47	0.0170	2.365	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
41	43.33	0.0170	2.328	29.6	202.4	0.0644	0.4096	225.9	0.453	248.0	0.490
42	44.18	0.0171	2.290	29.9	202.6	0.0651	0.4093	226.1	0.452	248.3	0.489
44	45.89	0.0171	2.216	30.7	203.0	0.0666	0.4087	226.6	0.451	248.8	0.489
46	47.61	0.0171	2.141	31.4	203.3	0.0680	0.4081	227.0	0.451	249.4	0.488
48	49.32	0.0172	2.067	32.2	203.7	0.0695	0.4075	227.5	0.450	249.9	0.488
50	51.03	0.0172	1.992	32.9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
52	53.00	0.0172	1.931	33.7	204.4	0.0724	0.4063	228.2	0.448	251.0	0.486
54	54.97	0.0173	1.870	34.4	204.7	0.0739	0.4056	228.6	0.448	251.5	0.486
56	56.94	0.0173	1.810	35.2	205.1	0.0754	0.4050	228.9	0.447	252.0	0.485
58	58.91	0.0173	1.749	35.9	205.4	0.0769	0.4043	229.3	0.447	252.5	0.485
60	60.88	0.0174	1.688	36.7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
62	63.13	0.0174	1.638	37.4	206.0	0.0798	0.4030	229.9	0.445	253.5	0.483
64	65.37	0.0174	1.588	38.2	206.3	0.0812	0.4024	230.3	0.444	254.0	0.483
66	67.62	0.0175	1.539	38.9	206.6	0.0827	0.4017	230.6	0.443	254.5	0.482
68	69.86	0.0175	1.489	39.7	206.9	0.0841	0.4011	231.0	0.442	255.0	0.482
70	72.11	0.0176	1.439	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
72	74.66	0.0176	1.398	41.1	207.5	0.0869	0.3998	231.6	0.440	256.0	0.480
74	77.21	0.0177	1.357	41.9	207.7	0.0883	0.3992	232.0	0.439	256.5	0.480
76	79.76	0.0177	1.315	42.6	208.0	0.0898	0.3985	232.3	0.439	256.9	0.479
78	82.31	0.0178	1.274	43.4	208.2	0.0912	0.3979	232.7	0.438	257.4	0.479
80	84.86	0.0178	1.233	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
82	87.74	0.0178	1.199	44.8	208.7	0.0940	0.3967	233.3	0.436	258.4	0.478
84	90.62	0.0179	1.165	45.6	209.0	0.0953	0.3960	233.6	0.435	258.9	0.477
86	93.50	0.0179	1.130	46.3	209.2	0.0967	0.3954	233.9	0.435	259.4	0.477
88	96.38	0.0180	1.096	47.1	209.5	0.0980	0.3947	234.2	0.434	259.9	0.476
90	99.26	0.0180	1.062	47.8	209.7	0.0994	0.3941	234.5	0.433	260.4	0.476
92	102.49	0.0180	1.033	48.6	209.9	0.1008	0.3935	234.8	0.433	260.8	0.476
94	105.72	0.0181	1.005	49.3	210.2	0.1022	0.3929	235.1	0.432	261.2	0.475
96	108.94	0.0181	0.9764	50.1	210.4	0.1035	0.3922	235.4	0.432	261.6	0.475
98	112.17	0.0182	0.9478	50.8	210.7	0.1049	0.3916	235.7	0.431	262.0	0.474
100	115.40	0.0182	0.9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
102	119.00	0.0183	0.8952	52.3	211.1	0.1076	0.3903	236.4	0.430	262.8	0.474

TABLE 4. PROPERTIES OF METHYL CHLORIDE—(Continued)

SAT. TEMP. F	ABS PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
104	122.60	0.0183	0.8712	53.1	211.3	0.1090	0.3897	236.8	0.430	263.2	0.473
106	126.20	0.0184	0.8471	53.8	211.4	0.1103	0.3890	237.1	0.429	263.5	0.473
108	129.80	0.0184	0.8231	54.6	211.6	0.1117	0.3884	237.5	0.429	263.9	0.472
110	133.40	0.0185	0.7990	55.3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
112	137.42	0.0185	0.7786	56.1	212.0	0.1144	0.3871	238.1	0.427	264.6	0.471
114	141.44	0.0185	0.7583	56.8	212.2	0.1157	0.3864	238.3	0.427	264.8	0.470
116	145.46	0.0186	0.7379	57.6	212.4	0.1171	0.3858	238.6	0.426	265.1	0.470
118	149.48	0.0186	0.7176	58.3	212.6	0.1184	0.3851	238.8	0.426	265.3	0.469
120	153.50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

is not known but it is stated that the action is brought about by surface condensation, and also by a difference between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures come into equilibrium. The amount of vapor adsorbed will depend on the adsorbent substances being used but for any single substance the amount depends on the temperature of the bed as well as on the partial pressure of the air-vapor mixture being passed over it.

As the process of adsorption goes on heat is liberated in the bed. The heat so liberated is the latent heat of the water vapor condensed together with the so-called heat of wetting. For a pound of water vapor at 60 F the latent heat released by condensation is approximately 1057 Btu. The heat of wetting for silica gel, for example, is about 200 Btu, making a total heat of adsorption of approximately 1257 Btu per pound of water adsorbed from the air-vapor mixture passing through the silica gel bed. The heat of wetting varies with the substance being used as the adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

Temperature-Pressure-Concentration Relations

Since the adsorptive ability of an adsorbent depends on the temperature of the bed and on the partial pressure difference between the pores and the air-vapor mixture it is important to know the pressures and temperatures at which pressure equilibrium is reached.

Evidently the equilibrium conditions represent the limits beyond which adsorption of vapor cannot continue. The relationship can be shown graphically and Fig. 1 is such a chart for silica gel. Charts of like nature can be plotted for other adsorbent materials.

Fig. 1 shows the equilibrium conditions for a gel bed maintained at constant temperature while the water vapor adsorption is allowed to continue until pressure equilibrium is reached. Each curve on the chart show a certain dew-point temperature, and therefore a certain pressure, of the saturated water vapor.

TABLE 5. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F_{11})

Sat. Temp. F	Abs Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

As an example in the interpretation of the chart consider the case when moist air at a temperature of 80 F and a partial vapor pressure of 0.5 in. of mercury flows through a bed of silica gel which is at a temperature of 80 F. The chart indicates that the equilibrium of pressure between the air-vapor mixture and the bed is reached when the dry bed has adsorbed moisture to the extent of 31 per cent of the weight when dry. When this happens the bed can adsorb no more moisture unless its temperature is changed.

In practice however the adsorbent bed is seldom held at a steady temperature in air conditioning applications and neither is the adsorption process permitted to continue until moisture equilibrium is reached. Instead, the bed temperature varies and the bed is re-activated before equilibrium is approached. While charts of this kind can show the limiting properties of the substances they are seldom directly applicable to the solution of air conditioning problems unless considerable additional information is available. This takes the form of performance data covering the characteristics of the equipment in which the adsorbent bed is placed. Such performance data are beyond the scope of this chapter.

Liquid Absorbents

Any absorbent substance may be used as an air drying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed. Absorbents are characteristically water solutions of materials in which the vapor pressure is

reduced to a suitable level by governing the concentration of the solution. In addition to having a suitable low vapor pressure, a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, chemically inert against any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

Water solutions, or brines, of the chlorides of various inorganic elements such as calcium chloride and lithium chloride are the absorbents most frequently used in connection with air conditioning applications and detailed attention is confined to these two in this chapter.

Nature of Absorption Process

The application consists of bringing the air-vapor stream into intimate contact with the absorbent, permissably by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a metal surface coil where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressures causes some of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent. In order that the process be continuous means must be provided for counteracting the diluting effect of the extracted moisture and also for maintaining the temperature of the brine sufficiently low to hold the desired vapor pressure.

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation which tends to raise the temperature of both the absorbent and the moist air stream. For every pound of water absorbed and condensed the heat added to the air stream and the brine

TABLE 6. PROPERTIES OF WATER

Sat. Temp. F	Abs Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From +32 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
32	0.0887	0.01602	3296.0	0.00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0.01602	2941.0	3.02	1074.4	0.0062	2.1724	1098.3	2.2172	1122.2	2.2581
40	0.1217	0.01602	2441.0	8.05	1076.8	0.0163	2.1555	1100.6	2.2000	1124.5	2.2406
45	0.1475	0.01602	2034.0	13.07	1079.2	0.0262	2.1390	1102.9	2.1832	1126.7	2.2234
50	0.1780	0.01602	1702.0	18.08	1081.5	0.0361	2.1230	1105.2	2.1667	1129.0	2.2066
55	0.2140	0.01603	1430.0	23.08	1083.9	0.0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206.0	28.08	1086.2	0.0556	2.0920	1109.8	2.1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0.0652	2.0771	1112.2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868.0	38.07	1090.9	0.0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43.06	1093.2	0.0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48.05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53.04	1097.8	0.1025	2.0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58.03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0.01612	404.2	63.01	1102.3	0.1206	1.9946	1125.6	2.0350	1148.9	2.0721
100	0.949	0.01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2.0588
105	1.101	0.01615	304.4	72.98	1106.8	0.1384	1.9695	1130.2	2.0093	1153.2	2.0458

For properties of steam at high temperatures, see Page 28.

TABLE 7. DEW-POINT OF AIR IN EQUILIBRIUM WITH LITHIUM CHLORIDE SOLUTIONS
CONCENTRATION IN GRAM MOLS OF LITHIUM CHLORIDE PER 1000 GRAINS WATER

0.0	2.0	4.0	6.0	8.0	10.0	12.0	14.0	16.0	18.0	20.0	22.0	24.0	26.0	28.0	30.0
320	315.2	308.7	299.9	290.2	279.7	269.4	259.6	251.5	244.1	236.5	230.0	223.8	218.6	214.5	210.3
300	295.4	289.1	280.5	270.9	260.6	250.5	240.8	232.6	225.4	218.0	211.8	205.8	200.8	196.9	192.8
280	275.6	269.5	261.1	251.7	241.5	231.6	222.2	214.0	206.7	199.7	193.5	187.8	183.2	179.3	175.2
260	255.8	250.0	241.9	232.6	222.7	212.8	203.5	195.5	188.4	181.7	175.4	170.0	165.6	162.0	158.4
240	236.0	230.4	222.5	213.5	203.8	194.2	185.0	177.1	170.0	163.6	157.5	152.2	148.3	144.6	140.5
220	216.2	210.8	203.2	194.4	184.9	175.5	166.4	158.6	151.6	145.3	139.6	134.6	130.7	127.3	124.2
200	196.4	191.2	183.9	175.4	166.1	156.7	148.0	140.3	133.5	127.3	121.9	117.0	113.3	110.1	
180	176.6	171.6	164.7	146.4	147.3	138.1	129.6	122.1	115.5	109.4	104.2	99.6	96.0		
160	156.8	152.1	145.4	137.4	128.6	119.7	111.3	103.9	97.4	91.6	86.6	82.2			
140	137.0	132.6	126.1	118.4	109.9	101.3	93.1	85.9	79.5	73.8	69.0				
120	117.2	113.0	106.8	99.4	91.1	82.7	74.7	67.8	61.5	56.0					
110	107.3	103.2	97.2	89.9	81.9	73.5	65.6	58.8	52.6	47.1					
100	97.4	93.4	87.5	80.5	72.7	64.4	56.6	49.8	43.7	38.2					
90	87.5	83.6	77.9	71.0	63.3	54.2	47.6	40.8	34.8	29.3					
80	77.6	73.8	68.4	61.6	54.0	46.1	38.5	31.8	25.9	20.6					
70	67.7	64.0	58.7	52.2	44.8	37.0	29.5	22.9	17.2	12.0					
60	57.8	54.3	49.1	42.7	35.5	27.9	20.5	14.0	8.3						
40	38.0	34.7	29.9	23.9	16.9	9.6	2.4	−3.9							
20		15.1	10.7	5.0	−1.7	−8.7	−15.4								
0		−4.5	−8.6	−13.9	−20.2	−27.0	−33.3								

combined is obtainable from steam tables. For instance, at 60 F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable.

Temperature-Pressure-Concentration Relations

Since the absorption process can continue only as long as there is a difference in vapor pressure between the absorbent and the air-vapor mixture and since at a given temperature of the absorbent the vapor pressure depends on the concentration of the solution, evidently there

TABLE 8. DENSITY OF LITHIUM CHLORIDE SOLUTIONS

CONCENTRATION MOLS LiCl PER 1000 GRAINS WATER	TEMPERATURE DEG F						
	0	50	100	150	200	250	300
0							
2		1.045	1.037	1.026	1.012		
4	1.090	1.085	1.076	1.064	1.052		
6	1.124	1.119	1.111	1.100	1.087		
8	1.156	1.150	1.143	1.132	1.122		
10	1.188	1.181	1.172	1.162	1.152	1.142	
12	1.217	1.209	1.199	1.188	1.178	1.168	
14	1.242	1.235	1.225	1.214	1.203	1.192	
16		1.257	1.248	1.236	1.226	1.215	
18		1.279	1.270	1.259	1.248	1.237	
20			1.291	1.280	1.279	1.568	
22				1.310	1.289	1.278	1.267
24				1.317	1.307	1.296	1.286
26					1.313	1.312	1.302
28					1.338	1.327	1.318
30						1.34	1.33
32							1.35

TABLE 9. VISCOSITY OF LITHIUM CHLORIDE SOLUTIONS (MILLIPOISE)

TEMP. DEG F	CONCENTRATION IN MOLAL												
	0	2	4	6	8	10	12	14	16	18	20	22	24
0			56.75	72.44	97.05	136.8	199.5						
20		28.91	37.07	47.42	63.09	84.94	123.3	178.6					
40	15.45	19.91	25.53	32.58	43.05	58.48	81.10	116.1	165.6				
60	11.02	14.26	18.37	23.55	30.90	41.40	56.62	79.80	111.2	156.3			
80	8.61	11.19	14.42	18.62	24.32	32.28	43.45	60.26	82.04	113.8			
100	6.82	8.89	11.48	14.94	19.36	25.59	33.96	46.13	61.52	84.72	118.3		
120	5.60	7.31	9.51	12.30	15.92	20.99	27.67	36.64	48.31	65.77	89.95		
140	4.70	6.15	8.07	10.42	13.43	17.66	22.96	30.06	38.99	52.48	71.12	95.94	
160	4.01	5.25	6.92	8.93	11.51	15.00	19.36	25.06	32.14	42.76	56.89	75.86	106.2
180	3.48	4.56	6.01	7.78	10.00	12.91	16.56	21.28	27.10	35.48	46.45	60.67	84.33
200	3.05	4.01	5.28	6.86	8.79	11.22	14.32	18.28	23.12	29.92	38.55	50.70	67.92
220	2.72	3.58	4.72	6.14	7.83	9.93	12.59	16.00	20.14	25.64	32.96	43.05	56.49
240	2.43	3.21	4.25	5.50	7.02	8.83	11.12	14.09	17.62	22.18	28.31	36.98	47.42
260	2.19	2.90	3.84	4.94	6.46	7.91	9.91	12.47	15.50	19.36	24.60	31.92	40.55
280	2.00	2.66	3.52	4.51	5.75	7.19	8.97	11.27	14.00	17.22	21.78	28.05	35.56
300	1.86	2.48	3.28	4.17	5.32	6.67	8.28	10.38	12.82	15.70	19.68	25.12	31.92
320	1.74	2.32	3.08	3.89	4.94	6.19	7.73	9.64	11.86	14.45	18.03	22.80	29.11

must be a relation between these quantities which if known would state the limits of the process. The relationship would also depend on the absorbent being used, and would have to be determined for each substance used as an absorbent. Fig. 2 shows this relationship graphically for lithium chloride. It will be noted that this chart is essentially similar to that shown in Fig. 1 and its direct usefulness is limited by much the same considerations.

In order to permit numerical calculations of air conditioning problems it is desirable to have tables for use instead of a chart like Fig. 2, and Tables 7, 8, 9 and 10 can be used in making calculations for lithium chloride.

TABLE 10. PROPERTIES OF LITHIUM CHLORIDE SOLUTIONS

CONCENTRATION MOLS (42.4 GRAMS *LiCl* PER 1000 GRAINS WATER)	PARTIAL HEAT OF MIXING AT 0 F BTU PER LB	TEMPERATURE COEF. OF PARTIAL HEAT OF MIXING BTU PER LB PER F	SPECIFIC HEAT AT 70 F	BOILING POINT F (AT 760 MM HG)	FREEZING POINT	SUBSTANCE THAT FIRST SEPARATES OUT ON FREEZING
0	0.0	0.0	0.998	212.0	32	Ice
2	2.04	−0.014	0.901	215.8	16.3	Ice
4	7.24	−0.036	0.831	221.5	−5.8	Ice
6	16.7	−0.069	0.778	228.9	−34.2	Ice
8	31.9	−0.109	0.739	238.1	−69	Ice
10	51.1	−0.143	0.710	248.4	−90	Ice
12	75.7	−0.160	0.687	258.8	−40	$LiCl\text{-}3H_2O$
14	90.8	−0.167	0.666	268.9	1	$LiCl\text{-}3H_2O$
16	124.8	−0.176	0.647	277.9	36.5	$LiCl\text{-}2H_2O$
18	145	−0.186	0.631	285.8	58.1	$LiCl\text{-}2H_2O$
20	162	−0.194	0.617	293.2	86.4	$LiCl\text{-}H_2O$
22	171	−0.20	0.604	300.2	133	$LiCl\text{-}H_2O$
24	177	−0.20	0.59	307	156	$LiCl\text{-}H_2O$
26	182	−0.21	0.58	313	180	$LiCl\text{-}H_2O$
28	191	−0.21	0.575	318	190	$LiCl\text{-}H_2O$
30	194	−0.21	0.57	323	195	$LiCl\text{-}H_2O$
32	198	−0.22	0.56	328	280	$LiCl$.

Instead of tabulating the vapor pressure of the solution of lithium chloride it is preferable to tabulate the dew-point of air in equilibrium with lithium chloride, since it is easy to interpolate between values of the dew-point and not so easy to interpolate accurately between values of vapor pressure. The values for dew-point may be converted to vapor

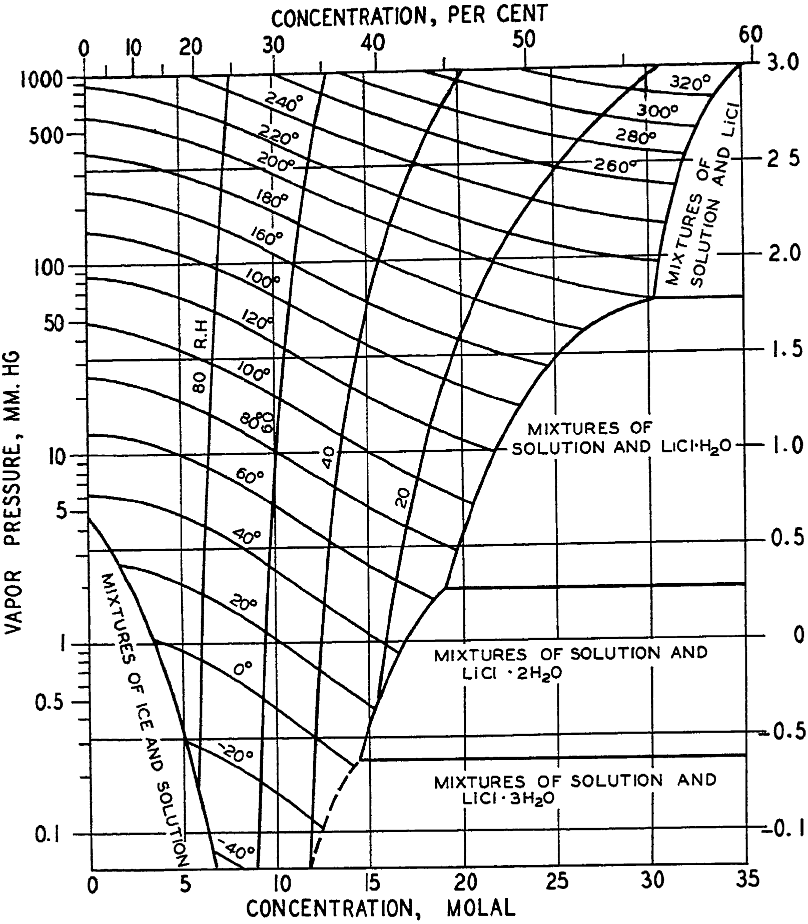

FIG. 2. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR LITHIUM CHLORIDE

pressures, relative humidity, and wet-bulb of air in equilibrium by means of the usual psychrometric chart or formula.

In Tables 7, 8, 9 and 10 the unit of concentration is the *mol.* A 'M' molal solution is defined as a solution containing $M \times 42.37$ grains of anhydrous lithium chloride per 1000 grains of water. The formula connecting concentration in mols with weight in per cent is equivalent to: $[100 \times M \times 42.37] \div [1000 + (M \times 42.37)]$.

PROBLEMS IN PRACTICE

1 ● What is the heat content above −40 F of 2.5 lb of ammonia when under a pressure of 92.9 lb per square inch gage and a temperature of 160 F?

First determine the condition of the ammonia at the stated temperature and pressure. Do this by finding the absolute pressure which in this case is 92.9 lb gage plus 14.7 or 107.6 lb per square inch absolute. From Table 1 note that the saturation temperature at this pressure is 60 F. Therefore, the ammonia is superheated 100 F, and the total heat per pound can be read directly from the Table as 689.9 Btu. In the 2.5 lb of ammonia there are 2.5 × 689.9, or 1724.75 Btu.

2 ● What volume is necessary to accommodate 0.27 lb of saturated F_{12} vapor when compressed to 99.6 lb gage?

The absolute pressure is 99.6 plus 14.7 or 114.3 lb. In Table 3 find that one pound of F_{12} vapor saturated occupies 0.368 cu ft. Then the 0.27 lb would occupy 0.27 × 0.368, or 0.099 cu ft.

3 ● How much heat would be removed from air passing over a coil through which 2 lb of methyl chloride per minute is forced? The coil is under a gage pressure of 64 lb per square inch and the liquid refrigerant is completely vaporized in passing through the coil.

Find that the absolute pressure is 64 plus 14.7 or 78.7 lb per sq in. From Table 4 note that the saturation temperature at this pressure is 75 F (nearly) and that the heat content per pound of the vapor is 207.8 Btu. Also that the heat content of the liquid is 42.2 Btu per pound. Subtract 42.2 from 207.8 and 165.6 Btu per pound is the heat necessary to change the liquid refrigerant to a vapor (latent heat). As the heat to accomplish this change comes from the air around the coil, the heat removed from the air is 165.6 Btu per pound of methyl chloride evaporated in the coil. When the refrigerant is circulated at 2 lb per minute, 2 × 165.6, or 331.2 Btu per minute are removed from the air, or refrigerating effect is produced at the rate of 331.2 ÷ 200, or 1.65 tons.

4 ● Calculate the dew-point, wet-bulb, relative humidity and absolute humidity of air in equilibrium at 100 F with pure lithium chloride solution of density 1.270.

From Table 8 the concentration of a solution of density 1.270 at 100 F is 18.0 *M*. From Table 7 the dew-point of 18 *M* lithium chloride at 100 F is 43.7 F. From Table 6, Chapter 1, the partial pressure of water over the solution is 0.2858 in. of Hg, the absolute humidity is 42.00 grains per pound dry air, and the wet-bulb is 65.8 F. The relative humidity is 14.0 per cent.

5 ● Calculate the boiling point, and freezing point of 18 M lithium chloride solutions.

From Table 10, boiling point (standard) is 285.8 F, freezing point is 58.1 F. The salt precipitated on cooling to this temperature has the composition $LiCl\text{-}2H_2O$.

6 ● Calculate the heat of vaporization of 1 lb of water from a large amount of lithium chloride solution at the boiling point.

The heat of boiling is equal to the heat of mixing plus the heat of boiling pure water at the same temperature. The heat of mixing from Table 10 at 18 *M* and 285.8 F is (145 − 0.186 × 285.8) = 92 Btu per pound. The heat of vaporization of water from steam tables at 285.8 F is 920 Btu per pound. Therefore the heat of vaporization of water from the solution is 920 + 92 = 1012 Btu per pound.

7 ● One thousand pounds of air per minute at 100 F dry-bulb with a dew-point of 70 F and a relative humidity of 39 per cent is passed over 18 *M* lithium chloride solution. The rate of flow of the solution is 200 gpm and the entering temperature is 80 F. The air leaves the absorber at 85 F dry-bulb and dew-point of 35 F. Calculate (*a*) the heat to be removed from the lithium chloride solution

to maintain these conditions, and (*b*) the temperature rise of the solution in passing through the absorber.

a. The heat content of the entering air: From Table 6, Chapter 1, weight of vapor at 70 F dew-point is 0.01574 lb times heat content of steam at 100 F dry-bulb is 1104.2 (Table 8, Chapter 1) equals 17.41 Btu per pound plus heat content of dry air at 100 F is 24.0 (Table 6, Chapter 1) resulting in heat content of mixture as 41.41 Btu per pound.

Similarly, the heat content of the leaving air: Weight of vapor at 35 F dew-point is $0.004262 \times 1097.5 = 4.68$ Btu per pound plus heat content of dry air at 85 F is 20.39 resulting in heat content of mixture as 25.07 Btu per pound.

Heat to be extracted from air is $1000 \times (41.41 - 25.07) = 16{,}340$ Btu per minute. Add to this the heat of mixing of 18*M* lithium chloride at 80 F equals $145 - (0.186 \times 80) = 130$ Btu per pound (Table 10) or for 1000 lb of air $\times (0.01574 - 0.00426) \times 130 = 1494$ Btu per minute. Heat to be removed from solution is $16{,}340 + 1494 = 17{,}834$ Btu per minute.

b. The weight of solution circulated is 200×1.275 (Table 8) $\times 8.33 = 2124$ lb per minute. Its heat capacity is 2124×0.631 (Table 10) $= 1340$ Btu per minute per degree Fahrenheit. The temperature rise is $17{,}834 \div 1340 = 13.31$ F.

Chapter 3

PHYSICAL AND PHYSIOLOGICAL PRINCIPLES OF AIR CONDITIONING

Vitiation of Air, Heat Regulation in Man, Effects of Heat, Effects of Cold and Temperature Changes, Acclimatization, Effective Temperature Index of Warmth, Optimum Air Conditions, Winter and Summer Comfort Zone, Optimum Humidity, Air Quality and Quantity, Air Movement and Distribution, Natural and Mechanical Ventilation, Heat and Moisture Losses, Ultra-Violet Radiation and Ionization, Recirculation and Ozone, Ventilation Standards

VENTILATION is defined in part as "the process of supplying or removing air by natural or mechanical means to or from any space." (See Chapter 45.) The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. More particularly, it includes the simultaneous control of temperature, humidity, movement, and purity of the air. The term is broad enough to embrace whatever other additional factors may be found desirable for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs. There is also a marked decrease in both positive and negative ions in the air of occupied rooms but the significance of this factor is still questionable[1].

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are much too small even under the worst conditions. The amount of carbon dioxide in air is

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[1]Changes in Ionic Content in Occupied Rooms Ventilated, by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 191).

often used in ventilation work as an index of odors of human origin, but the information it affords rarely justifies the labor involved in making the observation[2, 3]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired and transpired air is odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons, whether esthetic or physiological, call for the introduction of a certain minimum amount of clean outdoor air to dilute the odoriferous matter to a concentration which is not objectionable.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic bacteria from infected persons. Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as it was formerly believed. The large droplets do, of course, but minute droplets less than 0.1 mm. in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain infective organisms drift long distances with the air currents and the virus may remain alive long enough to be transmitted to other persons in the same room or building. Wells[4] recovered droplet nuclei from cultures of resistant micro-organisms a week after inoculation into a tight chamber of 300 cu ft capacity. Typical organisms of infections of the upper respiratory tract (pneumococcus type I, B. diphtheriae, Streptococcus hemolyticus, and Streptococcus viridans) were found to die out quite soon when exposed to light and air, and could be recovered from the air in small numbers only 48 hours after inoculation. Organisms typical of the intestinal tract (B. coli, B. typhosus, B. paratyphosus, A. and B. dysenteriae) were not recovered 12 hours after inoculation.

The significant factors in infection are believed to be the numbers of infective organisms encountered, the frequency of exposure, and the resistance of the individual including the degree of acquired immunity. The probability of encountering a sufficient number of organisms to break down the natural body defense is related to the air space per person and the quantity of clean air supplied. Except in badly ventilated rooms, the danger is believed to be "much contracted in space, limited in time and restricted to comparatively few diseases."[5]

Practical possibilities in sterilizing air supplies by the use of ultraviolet light are now being studied[6].

The primary factors in air conditioning work, in the absence of any specific contaminating source, are temperature, radiation, drafts and

[2]Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[3]Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. I. Coggins (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 133).

[4]Air-Borne Infection and Sanitary Air Control, by W. F. Wells, (*Journal Industrial Hygiene*, November, 1935).

[5]Preventive Medicine and Hygiene, by Milton J. Rosenau (6th edition, pp. 909-917, D. Appleton-Century Co., N. Y., 1935).

[6]Viability of B. Coli Exposed to Ultra-Violet Radiation in Air, by W. F. Wells, and G. M. Fair (*Science*, 1935, 82 p. 280).

body odors. As compared with these physical factors, the chemical factors are, as a general rule, of secondary importance.

HEAT REGULATION IN MAN

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends on the balance between heat production and heat loss. The heat resulting from the combustion of food within the body maintains the body temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, conduction and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. In healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to heat and cold. Nerve courses carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and little understood at the present time. Coordination of these different mechanisms seems to vary greatly with different air conditions.

With rising air temperatures up to 75 F or 80 F, metabolism, or internal heat production, decreases slightly[7], probably by an inhibitory action on heat producing organs, especially the adrenal glands, which seem to exert the major influence on basic combustion processes in the body. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, equally good results may be obtained by increasing the air movement, and hence the heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted, and the blood supply to the skin is curtailed by vasomotor shifts to the internal organs in order to conserve body heat. The sweat glands become inactive.

[7]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245).

EFFECTS OF HEAT

Although the human organism is capable of adapting itself to variations in environmental conditions, its ability to maintain heat equilibrium is limited. The upper limit of effective temperature to which the human organism is capable of adapting itself without serious discomfort or injury to health is 90 deg ET for men at rest and between 80 and 90 deg ET for men at work depending upon the rate of work. Within these limits a new equilibrium is established at a higher body temperature level through a chain of physiological adjustments. The heat regulating center fails, when the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 1.) In extreme heat, the metabolic rate is markedly increased owing to the excessive rise in body temperature[8], and a vicious cycle results which may eventually lead to serious physiologic damage.

TABLE 1. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp.	Actual Cheek Temp. (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp. (Deg Fahr per Hour)	Increase in Pulse Rate (Beats per Min per Hour)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp. (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt. by Perspiration (Lb per Hr)
60	-------	-------	-------	-------	225,000	0.0	6	0.5
70	-------	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	-------	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anæmic condition of the brain. The stomach loses some of its power to act upon the food, owing to a

[8]Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W. J. McConnell, C. P. Yaglou and W. B. Fulton (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 123).

diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[9]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather. The victim may lose appetite and suffer from indigestion, headache and general enervation, which may eventually lead to a premature old age.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss[10] recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride to a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show clearly that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

EFFECTS OF COLD AND TEMPERATURE CHANGES

The action of cold on human beings is not well known. Cold affects the human organism in two ways: (1) through its action on the body as a whole, and (2) through its action on the mucous membranes of the upper respiratory tract. Little exact information is available on the latter.

On exposure to cold, the loss of heat is increased considerably and only within certain limits is compensation possible by increased heat production and decreased peripheral circulation. The rectal temperature often rises upon exposure to cold but the pulse rate and skin temperature fall. The blood pressure increases, owing to constriction in the peripheral vessels. Just how cold affects health is not well understood. It imposes an extra load upon the heat-producing organs to maintain body temperature. The strain falls largely upon digestion, metabolism, blood circulation, and the kidneys, and indirectly upon the nervous system[11].

Although the seasonal increase in morbidity and mortality sets in with the approach of cold weather and subsides in the warm summer months, little is known of the specific causative factors and their mechanism of action. Over-crowding of buildings, overheated rooms, lack of ventilation, and close personal contacts are frequently held responsible for our winter ills, but the evidence is not conclusive.

In extremely cold atmospheres compensation by increased metabolism becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

[9]Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.* Vol. 24, 1927, p. 832).

[10]Some Effects of High Air Temperatures upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284).

[11]Loc. Cit. Note 5.

Cannon showed that excessive loss of heat is associated with increased activity of the adrenal medulla[12]. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast[13] found a degeneration of thyroid and adrenal glands upon exposure to cold.

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced in passing from a warm room to the cold air out-of-doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils, throat, and nasal mucosa, which is accompanied by a fall in the temperature of the tissues. On re-warming, the palate and throat do not always regain their normal temperature and blood supply. This anæmic condition favors bacterial activity and it probably plays a part in the disposition of common cold and other respiratory diseases. It is believed that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England[14] found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and general laborers, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries[15, 16]. According to these data the highest pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue.

[12]Studies on the Condition of Activity of Endocrine Glands, by W. B. Cannon, A. Guerido, S. W. Britton and E. M. Bright (*American Journal of Physiology*, Vol. 79, 1926, p. 466).

[13]Studies in Exhaustion Due to Lack of Sleep, by T. H. Bast, J. S. Supernaw, B. Lieberman and J. Munro (*American Journal of Physiology*, Vol. 85, 1928, p. 135).

[14]Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (*Industrial Fatigue Research Board*, Report No. 5, 1920, London).

[15]Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin*, Metropolitan Life Insurance Company, 1928).

[16]The Pneumonia Problem in the Steel Industry, by D. K. Brundage and J. J. Bloomfield, (*Journal of Industrial Hygiene*, 14, December, 1932).

They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of the present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to an environment averaging between 60 and 80 F is a very primitive trait[17].

Within these limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat have a lower rate of heat production than do those who grow up in cooler environments. This causes them to stand chilling poorly as they are unable to quickly increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adrenal activity has slowed down. Within a couple of years, however, they find themselves standing the heat much better and disliking cold. They become acclimated by a definite change in the combustion level within the body[18].

In certain individuals the psychologic factor is more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed regardless of the quality of the air. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

EFFECTIVE TEMPERATURE INDEX OF WARMTH

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler.

On the other hand, in cold environments an increase in humidity produces a cooler sensation. The dividing line at which humidity has no effect upon warmth varies with the air velocity and is about 46 F (dry-bulb) for still air and about 51, 56 and 59 F for air velocities of 100, 300 and 500 fpm, respectively. Radiation from cold or warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity and air movement which induce the same feeling of warmth are called thermo-equivalent con-

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[17]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1924.

[18]Air Conditioning in its Relation to Human Welfare, by C. A. Mills (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 289).

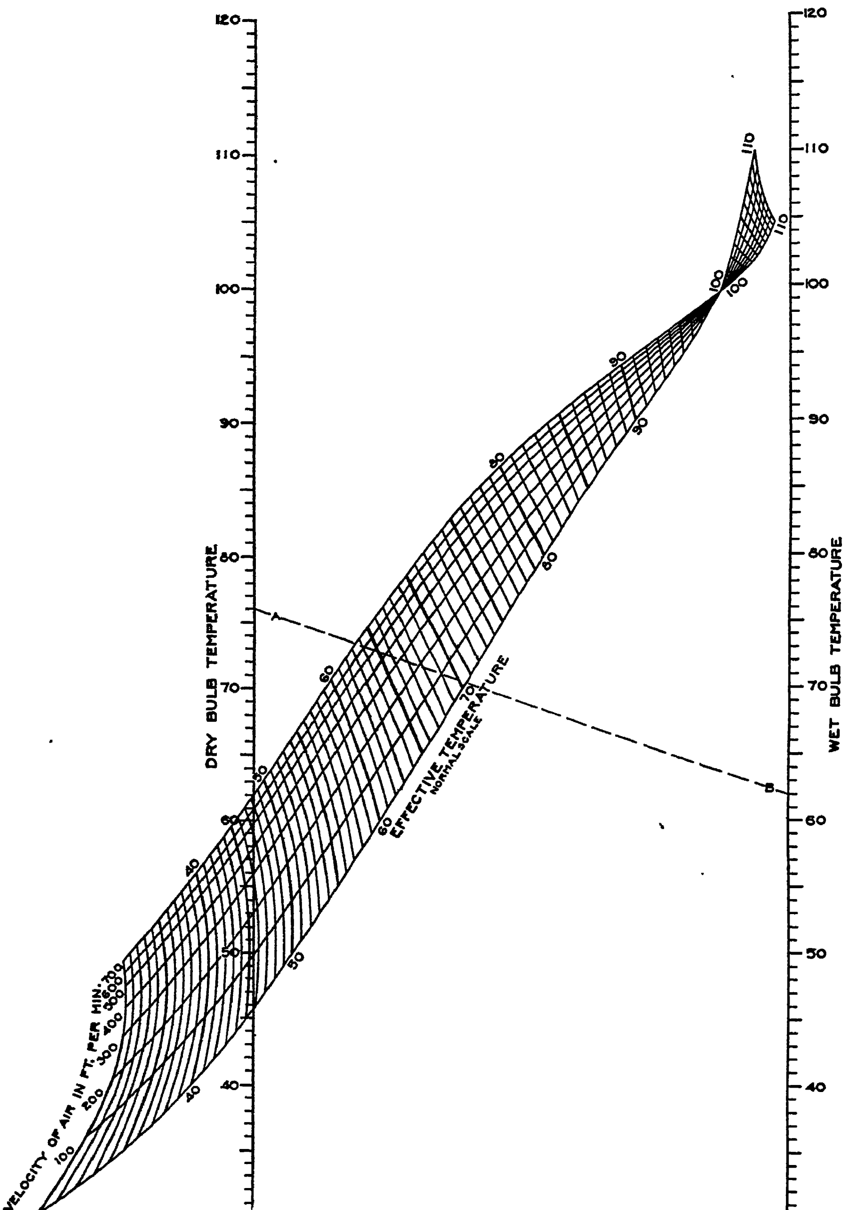

FIG. 1. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.* warm air, direct steam or hot water radiators, plenum systems.

ditions. A series of tests[19, 20, 21, 22] at the A.S.H.V.E. Research Laboratory, Pittsburgh, established the equivalent conditions met with in general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

Effective temperature is not in itself an index of comfort, except under ordinary humidity conditions (30 to 60 per cent) when the individual is least conscious of humidity. Moist air at a comparatively low temperature, and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be different, although the effective temperature is the same.

Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discomfort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture.

The numerical value of the effective temperature index for any given air condition is fixed by the temperature of calm (15 to 25 fpm air movement) and saturated air which induces a sensation of warmth or cold like that of the given condition. Thus, any air condition has an effective temperature of 60 deg, for instance, when it induces a sensation of warmth like that experienced in calm air at 60 deg saturated with moisture. The effective temperature index cannot be measured directly but it is computed from the dry- and wet-bulb temperature and the velocity of air using charts (see Figs. 1 or 2, 3 and 4) or tables. The accuracy in estimating effective temperature is ± 0.5 F, because the human organism cannot perceive smaller temperature differences. Therefore, there is no justification in trying to read chart values closer than 0.5 F, as this implies fictitious accuracy.

The charts shown in Figs. 1, 2, 3 and 4 apply to average normal and healthy persons adapted to American living and working conditions. Application is limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, central fan and direct hot water and steam heating systems) in which the difference between the air and wall surface temperatures may not be too

[19]Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

[20]Cooling Effect on Human Beings by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193).

[21]Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89).

[22]Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 315).

great. The charts do not apply to rooms heated by radiant method such as British panel system, open coal fires and similar usages. They will probably not apply to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different than those of the subjects employed in experiments at the Research Laboratory.

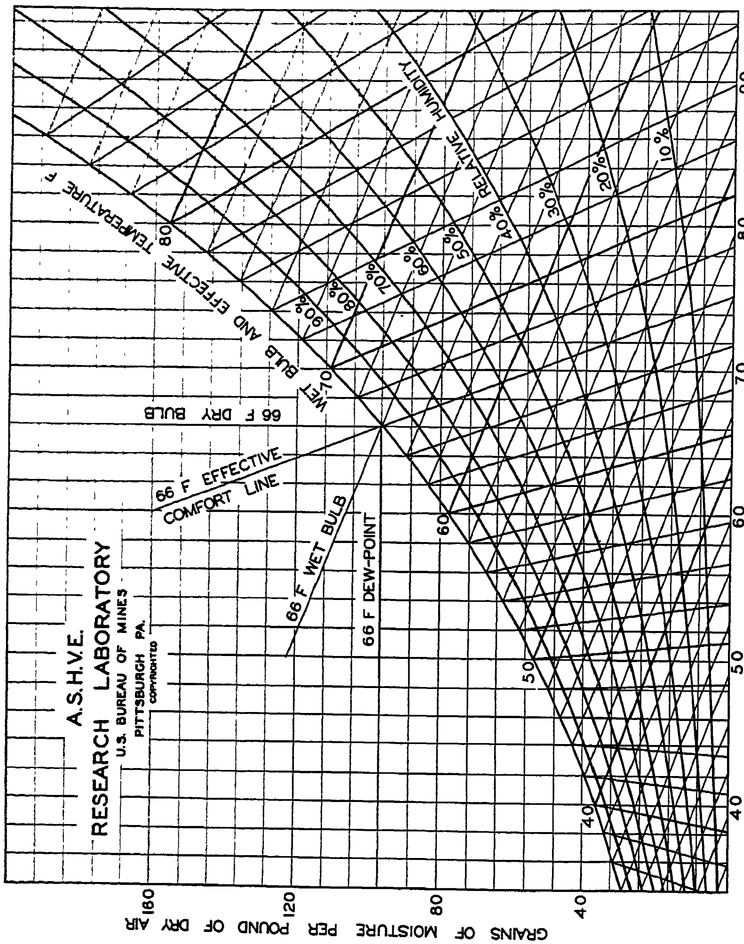

FIG. 2. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

In rooms in which the average wall surface temperature is considerably below or above air temperature, a correction must be applied to the readings of the dry-bulb thermometer to allow for such negative or positive radiation. In Fig. 5 is given the cooling effect of cold walls as determined at the A.S.H.V.E. Research Laboratory[23] by trained subjects

[23]Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 83).

passing back and forth from a small experimental room having three cold walls, to a control room with walls and air at the same temperature.

It can be seen in Fig. 5 that with air and walls at 70 F in the control (warm wall room), the cooling effect of three cold walls at 55 F of the experimental room was 4 F. Therefore, for the same feeling of warmth,

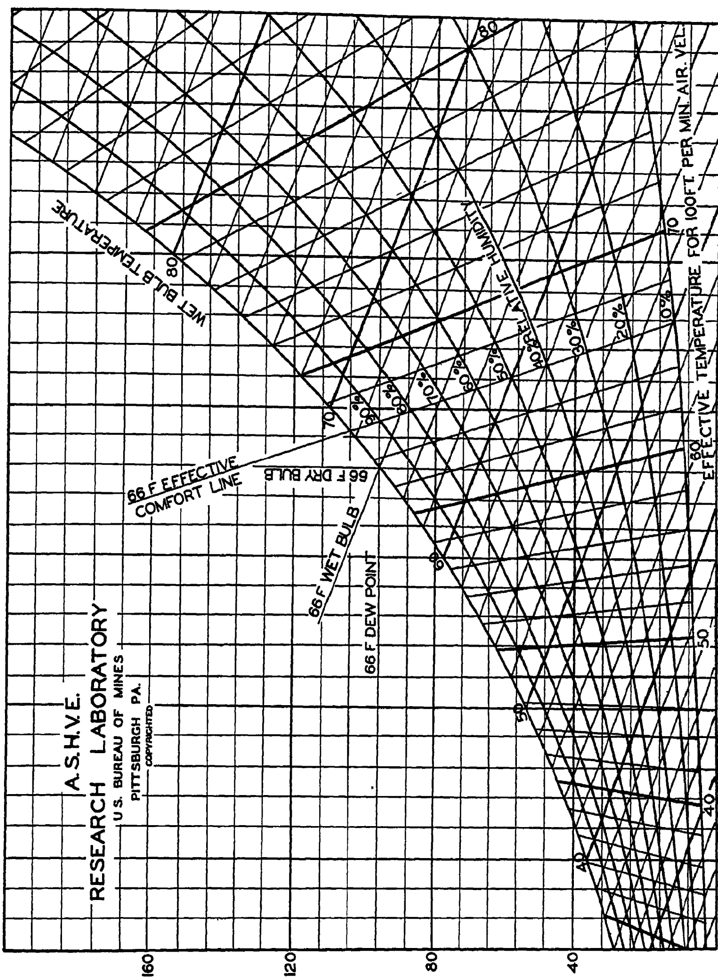

FIG. 3. PSYCHROMETRIC CHART, PERSONS NORMALLY CLOTHED AT REST, IN 100 FPM AIR VELOCITY

the temperature in the experimental room should be increased to 74 F. The reverse would hold in rooms with high-wall surface temperature; a lower air temperature would be required to compensate for positive radiations to the occupants.

OPTIMUM AIR CONDITIONS

No single comfort standard can be laid down which would meet every need. There is an inherent individual variation in the sensation of warmth or comfort felt by persons when exposed to an identical atmospheric condition. The state of health, age, sex, clothing, activity, and

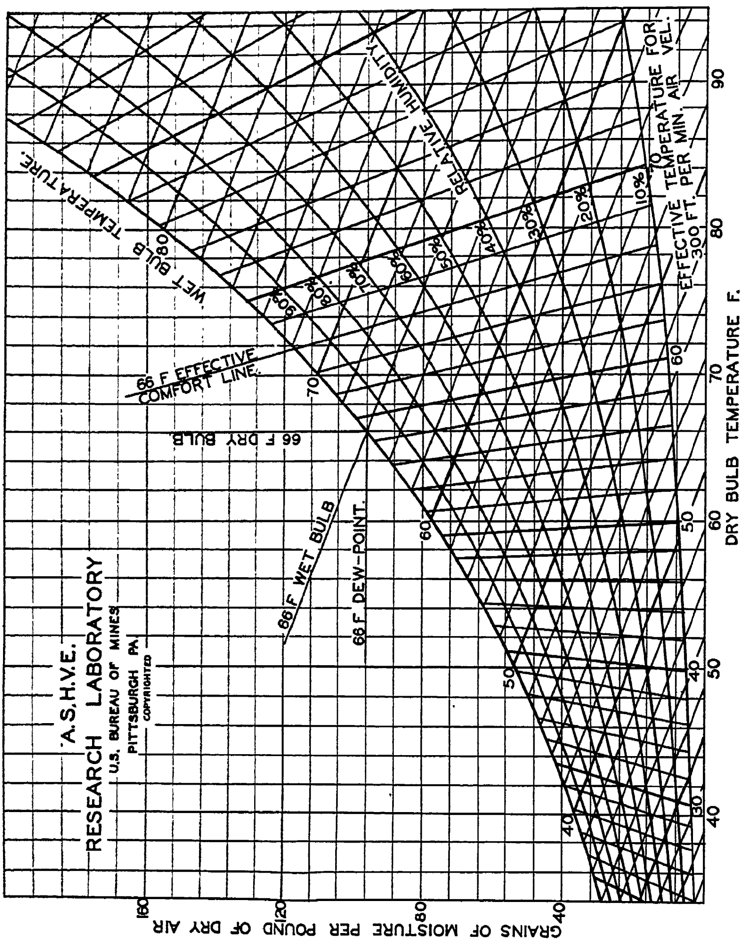

FIG. 4. PSYCHROMETRIC CHART, PERSONS NORMALLY CLOTHED AT REST, IN 300 FPM AIR VELOCITY

the degree of acquired adaptation seem to be the important factors affecting the comfort standards.

Since the prolonged effects of temperature, humidity and air movement on health are not known to the same extent as their effects on comfort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, the two do not differ greatly since this is in accordance with the efficient operation of the

heat regulating mechanism of the body. This belief is strengthened by results of studies on premature infants over a four-year period[24]. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhœa and mortality was decreased, gains in body weight increased and infections were reduced to a minimum.

Winter Comfort Zone and Comfort Line

In Fig. 6 is shown the A.S.H.V.E. winter comfort zone which was determined experimentally with large groups of men and women subjects wearing customary indoor winter clothing. The extreme comfort zone includes conditions between 60 and 74 deg ET in which one or more of the experimental subjects were comfortable. The average comfort zone

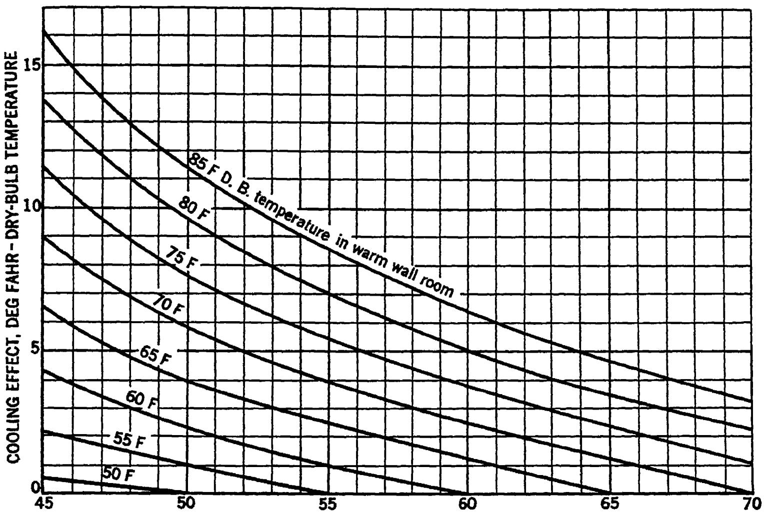

FIG. 5. COOLING EFFECT OF THREE COLD WALLS IN A SMALL EXPERIMENTAL ROOM, AS DETERMINED BY COMPARISON WITH SENSATIONS IN A ROOM OF UNIFORM WALL AND AIR TEMPERATURE

includes conditions between 63 and 71 deg ET conducive to comfort in 50 per cent or more of the experimental subjects. The most popular effective temperature was found to be 66 deg, and was adopted by the Society[25] as the *winter comfort* line for individuals at rest wearing customary winter clothing.

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal

[24]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 383).

[25]How to Use the Effective Temperature Index and Comfort Charts (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 410).

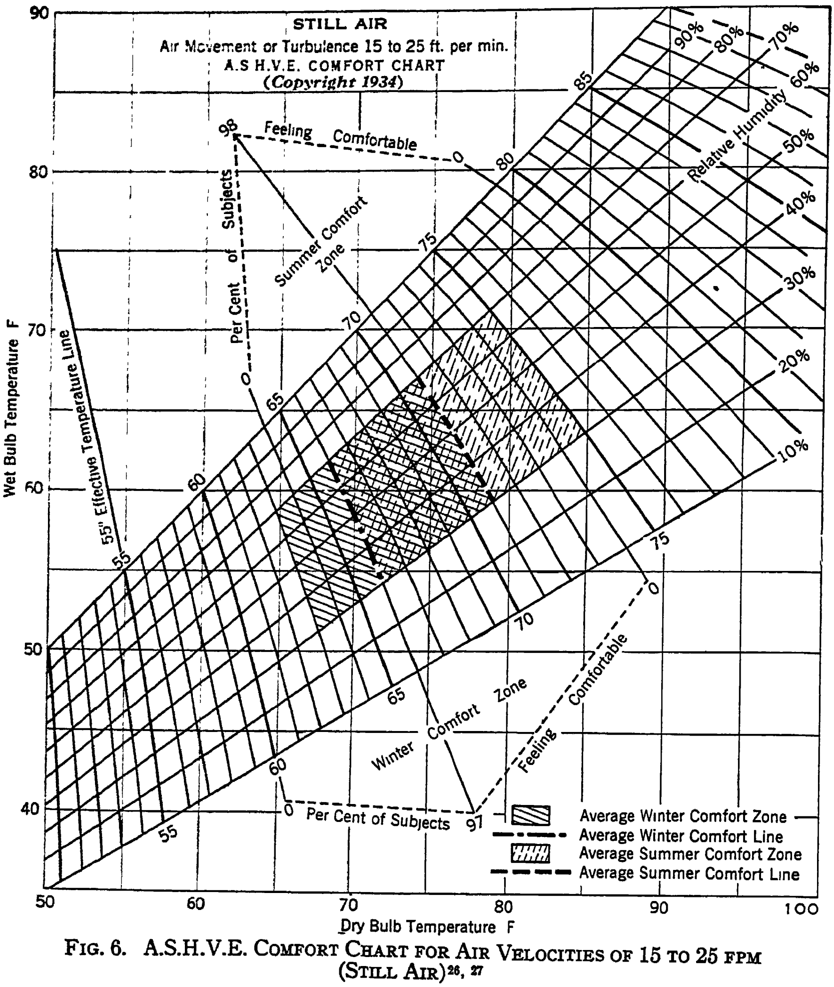

FIG. 6. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)[26, 27]

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

equilibrium with its environment with the least conscious sensation to the individual, or with the minimum phsyiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity, but also the condition of the surrounding objects and wall surfaces. The comfort zone tests were

[26]Determination of the Comfort Zone With Further Verification of Effective Temperatures Within This Zone, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

[27]The Summer Comfort Zone; Climate and Clothing, by C. P. Yaglou and Philip Drinker (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 269).

made in rooms with wall surface temperatures approximately the same as the room dry-bulb temperature. For walls of large area having unusually high or low surface temperatures, however, a somewhat lower or higher range of effective temperature is required to compensate for the increased gain or loss of heat to or from the body by radiation as shown in Fig. 5. (See also Chapter 41).

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, rooms with excessive glass area or rooms with poorly insulated or cold walls. Even in the warm south and southwestern climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures may be necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants in close proximity[28].

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm*. These individual differences among the minority should be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

The comfort chart (Fig. 6) applies to adults between 20 and 70 years of age living in the northeastern parts of the United States. For prematurely born infants, the optimum temperature varies from 100 F to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[29]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 F to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 F to 68 F temperature with a moderate humidity (not specified) and a moderate but not excessive amount of air movement (not specified)[30].

[28]Loc. Cit. Note 27.

[29]Loc. Cit. Note 24.

[30]Ventilation (Report N. Y. State Commission on Ventilation. E. P. Dutton and Co., N. Y., 1923).

Satisfactory comfort conditions for men at work are found to vary from 40 deg to 70 deg ET, depending upon the rate of work and amount of clothing worn[31]. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

Summer Comfort Zones

The summer comfort zone is much more difficult to fix than the winter zone owing to the complicating factor of sweating in warm weather. A given air condition which is comfortable for persons with dry skin and clothing may prove too cold for those perspiring, as is the case, for instance, with employees and customers in a cooled store, restaurant, or theater, on a warm summer day. The conditions to be maintained in different types of public buildings depend to a large extent upon the occupants' length of stay and upon the prevailing outdoor condition.

In Fig. 6 is shown the summer comfort zone for exposures of 3 hours or more, after adaptation has taken place. The average zone extends from 66 to 75 deg ET, with a comfort line at 71 deg ET, as determined at the Harvard School of Public Health[32]. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

The best effective temperature (for exposures lasting 3 hours or more) was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approximately the same value in July, August and September, and although the average monthly temperature did not vary much, the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Crowding the experimental chamber lowered the comfortable effective temperature from 70.8 deg when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft, to 69.4 deg when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

The basic summer comfort zone, shown in Fig. 6 has more academic than practical significance. It prescribes conditions of choice for continuous exposures, as in homes, offices, etc., without regard to costs, prevailing outdoor air conditions, and temperature contrasts upon entering or leaving the cooled space. A great number of persons seem to be content with a higher plane of indoor temperature, particularly when the matter of first cost and cost of operation of the cooling plant is given due consideration.

According to previous investigations[33], an indoor temperature of about 80 F with relative humidities below 55 per cent, or 74.5 deg ET and lower, result in satisfactory comfort conditions in the living quarters of a residence,

[31]Loc. Cit. Note 22.
[32]Loc. Cit. Note 27.
[33]Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207).

and while this condition is not representative of optimum comfort it provides for sufficient relief in hot weather to be acceptable to the majority of users. Experience in a number of air conditioned office buildings, including the New Metropolitan Life Building in New York[34], indicates that a temperature of about 80 F with a relative humidity between 45 and 55 per cent (73 to 74.5 deg ET) is generally satisfactory in meeting the requirements of the employees.

In artificially cooled theaters, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is to provide sufficient relief from the heat without causing sensations of chill or intense heat on entering and leaving the building.

Effective temperatures as high as 75 F at times have been found satisfactory in very warm weather. There are two schools of thought concerning the relation between temperature and humidity to be maintained. For a given effective temperature some engineers including the operators of cooling plants favor a comparatively low temperature with a high humidity as this results in a reduction of refrigeration requirements. Preliminary experiments at the A.S.H.V.E. Laboratories[35] would seem to indicate no appreciable impairment of comfort with relative humidities as high as 80 per cent, provided the effective temperature is between 70 and 75 deg.

The second school favors a higher dry-bulb temperature, according to the prevailing outdoor dry-bulb, with a comparatively low humidity (well below 50 per cent); the main purpose being to reduce temperature contrasts upon entering and leaving the *cooled* space and to keep the clothing and skin dry. This second scheme requires more refrigeration with the present conventional type of apparatus.

Current practice in theatres, restaurants, etc., follows a schedule similar to that shown in Table 2. This schedule should be used with considerable judgment depending on the occupancy and local climatic conditions. There are some indications that a definite indoor effective temperature may be applicable throughout the cooling season, but other observations seem to show that changing indoor conditions are desirable with violently changing outdoor weather conditions. It is, in fact, questionable whether entirely satisfactory air conditions could be adduced for practical use to meet the changing requirements of patrons from the time they enter to the time they leave a cooled space. Too many uncontrollable variables enter into the problem. Work now going on at the A.S.H.V.E. Laboratories and other interested institutions may throw considerable light on this complex problem.

For cooled banks and stores where the customers come and go spending

[34]The Air Conditioned System of the New Metropolitan Building—First Summer's Experience, by W. J. McConnell and I. B. Kagey (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 217).

[35]Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215). Cooling Requirements for Summer Air Conditioning, by F. C. Houghten, F. E. Giesecke, Cyril Tasker and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936, p. 681).

TABLE 2. DESIRABLE INSIDE CONDITIONS IN SUMMER CORRESPONDING TO OUTSIDE TEMPERATURES[a]

Occupancy Over 40 Min

Outside Dry-Bulb Deg F	Inside Air Conditions: Effective Temperature	Dry-Bulb Deg F	Wet-Bulb Deg F	Dew-Point Deg F	Relative Humidity Per Cent
100	75	83	66	56	40
	75	82	67	59	45
	75	81	68	61	51
	75	80	70	65	60
95	74	82	64	53	36
	74	81	66	57	44
	74	80	67	60	51
	74	79	68	62	57
	74	78	70	66	68
90	73	81	63	52	36
	73	80	64	54	41
	73	79	66	59	50
	73	78	67	61	56
85	72	80	61	48	32
	72	79	63	53	41
	72	78	64	56	46
	72	77	66	60	56
80	71	78	61	49	36
	71	77	63	54	45
	71	76	64	57	52
	71	75	66	61	61

[a]Applicable to individuals engaged in sedentary or light muscular activity.

but a few minutes in the cooled space, observations[36] indicate a schedule about 1 deg dry-bulb or effective temperature higher than that shown in Table 2. Laboratory experiments with exposures of 2 to 10 min indicate temperatures 2 to 10 F higher than those in Table 2 but with much lower relative humidities.

It should be kept in mind that southern people, with their more sluggish heat production and lack of adaptability, will demand a comfort zone several degrees higher than that for the more active people of northern climates. Instead of the summer comfort line standing at 71 deg as here given, it was found to be much higher for foreigners in Shanghai where climatic conditions are similar to those of our gulf states. This difference in adaptability of people forms a very real problem for air conditioning engineers. Cooling of theaters, restaurants, and other public buildings in southern climates cannot be based on northern standards without considerable modification.

Optimum Humidity

Just what the optimum range of humidity is, is a matter of conjecture. There seems to exist a general opinion, supported by some experimental and statistical data, that warm, dry air is less pleasant than air of a

[36]How Cool? Inside Temperature should Depend upon Type of Occupancy, by J. H. Walker (*Heating and Ventilating*, October, 1932).

moderate humidity, and that it dries up the mucous membranes in such a way as to increase susceptibility to colds and other respiratory disorders [37, 38, 39]. Owing to the cooling effect of evaporation, higher temperatures are necessary, and this condition may lead to discomfort and lassitude. Moist air, on the other hand, interferes with the normal evaporation of moisture from the skin, and again may cause a feeling of oppression and lassitude, especially when the temperature is also high. For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[40] until the infants reach a weight of about 5 lb. No such clear-cut evidence exists in the case of adult persons. In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 per cent and 60 per cent with ordinary room temperatures. This is in accord with studies by Howell[41], Miura[42] and others.

The limitation of the comfort zones in Fig. 6 with respect to humidity must not be taken too seriously. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort, so long as extremely low humidities are avoided. From the standpoint of health, however, the consensus seems to favor a relative humidity between 40 and 60 per cent. In mild weather such comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken to properly insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. Information on the prevention of condensation on building surfaces is given in Chapter 7. The principles underlying humidity requirements and limitations are discussed more fully elsewhere[43].

The purpose of artificial humidification may be easily defeated by failure to change the spray water of the humidifier at least daily. Where this condition occurs, the air is characterized by a *lack of freshness*, and under extreme conditions by a musty, sour odor in the conditioned space.

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[37]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al (*Journal Experimental Medicine*, 1921, Vol. 34, p. 11).

[38]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A. Goldman, et al and Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, Vol. 29, p. 151).

[39]The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al (*Am. Otol., Rhinol., and Laryngol.*, 1921).

[40]Loc. Cit. Note 24.

[41]Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).

[42]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

[43]Humidification for Residences, by A. P. Kratz, University of Illinois (*Engineering Experiment Station Bulletin* No. 230, July 28, 1931).

AIR QUALITY AND QUANTITY

Air Quality

In occupied spaces in which the vitiation is entirely of human origin, the chemical composition of the air, the dust, and often the bacteria content may be dismissed from consideration so that the problem consists in maintaining a suitable temperature with a moderate humidity, and in keeping the atmosphere free from objectionable odors. Such unpleasant odors, human or otherwise, can be easily detected by persons entering the room from clean, odorless air.

In industrial rooms where the primary consideration is the control of air pollution (dusts, fumes, gases, etc.), or contamination not removable at the source of production, the clean air supply must be sufficient to dilute the polluting elements to a concentration below the physiological threshold (see Chapters 4 and 26).

Air Quantity

The air supply to occupied spaces must always be adequate to satisfy the physiological requirements of the occupants. It must be sufficient to maintain the desired temperature, humidity, and purity with reasonable uniformity and without drafts. In many practical instances there are two air quantities to be considered, (a) outdoor air supply, and (b) total air supply. The difference between the two gives the amount of air to be recirculated.

When the only source of contamination is the occupant, the minimum quantity of outdoor air needed appears to be that necessary to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including socio-economic status of occupants, outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, temperature, and other factors of secondary importance. With any given group of occupants and type of air conditioner the intensity of body odor perceived upon entering a room from relatively clean air was found to vary inversely with the logarithm of outdoor air supply and the logarithm of the air space allowed per person.

The minimum outdoor air supply necessary to remove objectionable body odors under various conditions, as determined experimentally at the Harvard School of Public Health[44], is given in Table 3.

Outdoor air requirements for the removal of objectionable tobacco smoke odors have yet to be determined. Practical values in the field vary from 5 to 15 cfm per person; this air quantity may and should be a part of that necessary for other requirements, *i.e.*, removal of body odors, heat, moisture, etc.

The total quantity of air to be circulated through an enclosure is governed largely by the needs for controlling temperature and air distribution when either heating or cooling is required. The factors which determine total air quantity include the type and nature of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

Serious difficulties are often encountered in attempting to cool a room

[44]Loc. Cit. Note 3.

with a poor distribution system or with an air supply which is too small to result in uniform distribution without drafts. Some systems of distribution produce drafts with but a few degrees temperature rise, while other systems operate successfully with a temperature rise as high as 35 F. The total air quantity introduced in any particular case is inversely proportional to the temperature rise, and depends largely upon the judgment and ingenuity of the engineer in designing the most suitable system for the particular conditions.

TABLE 3. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS

(Provisional values subject to revision upon completion of work)

TYPE OF OCCUPANTS	AIR SPACE PER PERSON CU FT	OUTDOOR[a] AIR SUPPLY CFM PER PERSON
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

[a]Impressions upon entering room from relatively clean air at threshold odor intensity.

The changes in moisture content resulting from occupation in the atmosphere of a room supplied with various volumes of outside air is shown in Fig. 7. Data are given for an adult, 5 ft 8 in. in height, weighing 150 lb and having a body surface of 19.5 sq ft and for a child, 12 years of age, 4 ft 7 in. in height weighing 76.6 lb and having a body surface area of 12.6 sq ft. Also given in Fig. 7 is the temperature of incoming air necessary to maintain a room temperature of either 70 or 80 F as indicated assuming that there is no heat gain or loss to the room by transmission through the walls, solar radiation or other sources.

AIR MOVEMENT AND DISTRIBUTION

Stagnant warm air, no matter how pure, is not stimulating and it detracts to some extent from the quality of air. Experience, and recent field studies by the A.S.H.V.E. Research Laboratory[45] place the desirable air movement between 15 and 25 fpm under ordinary room temperatures during the heating season. Objectionable drafts are likely to occur when the velocity of the air current is 40 fpm and the temperature of the air current 2 F or more below the customary winter room temperature. Higher velocities are not objectionable in the summer time when the air

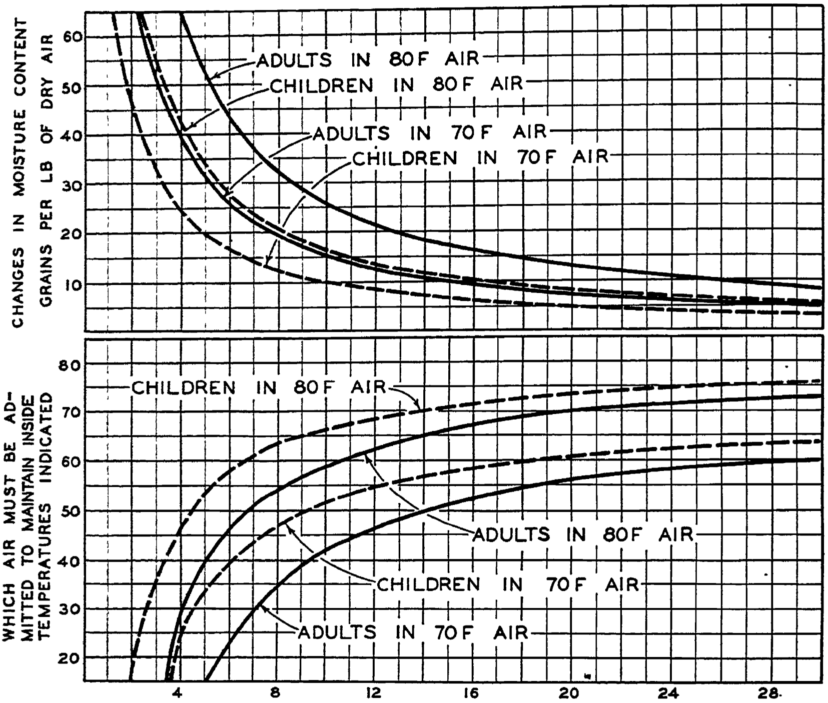

FIG. 7. RELATION AMONG RATE OF AIR CHANGE PER OCCUPANT, MOISTURE CONTENT OF ENCLOSURE, AND DRY-BULB TEMPERATURE OF INCOMING AIR

temperature exceeds 80 F. Variations in air movement and temperature in different parts of occupied rooms are often indicative of relative air distribution. The work of the A.S.H.V.E. Research Laboratory indicates that an air movement between 15 and 25 fpm with a temperature variation of 3 F or less in different parts of a room, 36 in. above floor, insure satisfactory distribution. Considerable evidence was obtained in these tests to show that measurements of carbon dioxide are not essential for the study of air distribution, or for indirect measurements of outdoor air

[45]Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 268).

TABLE 4. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[a]

Activity	Hourly Metabolic Rate for Avg. Person or Total Heat Dissipated, Btu per Hour	Hourly Sensible Heat Dissipated, Btu per Hour	Hourly Latent Heat Dissipated, Btu per Hour	Moisture Dissipated, per Hour	
				Grains	Lb
Average Person Seated at Rest[1]	384	225	159	1070	0.153
Average Person Standing at Rest[1]	431	225	206	1390	0.199
Tailor[2]	482	225	257	1740	0.248
Office Worker Moderately Active	490	225	265	1790	0.256
Clerk, Moderately Active, Standing at Counter	600	225	375	2530	0.362
Book Binder[2]	626	225	401	2710	0.387
Shoe Maker[2]; Clerk, Very Active Standing at Counter	661	225	436	2940	0.420
Pool Player	680	230	450	3040	0.434
Walking 2 mph[3, 4]; Light Dancing	761	250	511	3450	0.493
Metal Worker[2]	862	277	585	3950	0.564
Painter of Furniture[2]	876	280	596	4020	0.575
Restaurant Serving, Very Busy	1000	325	675	4560	0.651
Walking 3 mph[3]	1050	346	704	4750	0.679
Walking 4 mph[3, 4]; Active Dancing, Roller Skating	1390	452	938	6330	0.904
Stone Mason[2]	1490	490	1000	6750	0.964
Bowling	1500	490	1010	6820	0.974
Man Sawing Wood[2]	1800	590	1210	8170	1.167
Slow Run[4]	2290	.....	.....	.....	.....
Walking 5 mph[3]	2330	.....	.....	.....	.....
Very Severe Exercise[5]	2560	.....	.....	.....	.....
Maximum Exertion Different People[4]	3000 to 4800	.....	.....	.....	.....

[a]Metabolism rates noted based on tests actually determined from the following authoritative sources: [1]A.S.H.V.E. Research Laboratory; [2]Becker and Hamalainen; [3]Douglas, Haldane, Henderson and Schneider; [4]Henderson and Haggard; and [5]Benedict and Carpenter. Metabolic rates for other activities estimated. Total heat dissipation integrated into latent and sensible rates by actual tests for metabolic rates up to 1250 Btu per hour, and extrapolated above this rate. Values for total heat dissipation apply for all atmospheric conditions in a temperature range from approximately 60 to 90 F dry-bulb. Division of total heat dissipation rates into sensible and latent heat holds only for conditions having a dry-bulb temperature of 79 F. For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true.

TABLE 5. DEGREES OF PERSPIRATION FOR PERSONS SEATED AT REST UNDER VARIOUS ATMOSPHERIC CONDITIONS

Degree of Perspiration[a]	Atmospheric Condition					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B.
Forehead clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body damp	**79.0**	79.7	78.4	**81.0**	97.5	67.5
Beads on forehead	**80.0**	80.8	79.4	**87.0**	109.4	75.2
Body wet	**84.5**	85.4	84.0	**86.5**	108.5	74.6
Perspiration on forehead runs and drips	**88.0**	89.0	87.6	**94.0**	125.2	85.4
Perspiration runs down body	**88.5**	89.5	88.1	**90.0**	116.0	79.5

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

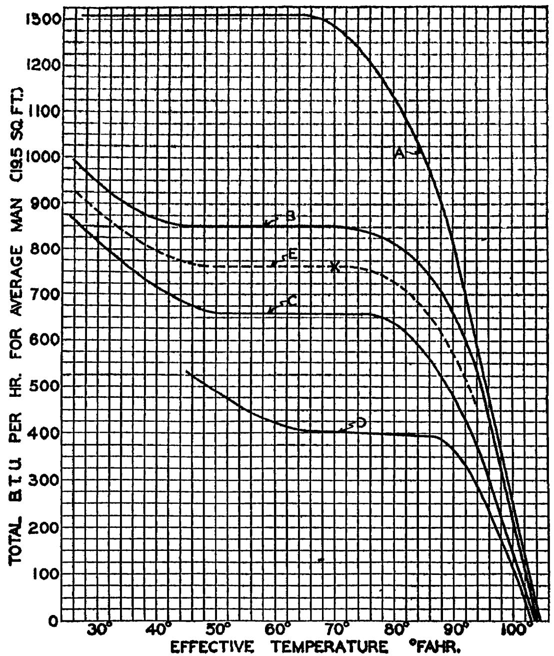

FIG. 8. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,160 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at an effective temperature of 70 deg only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

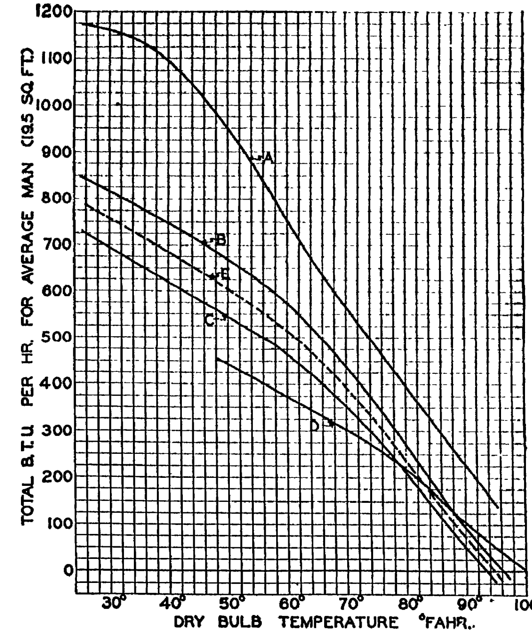

FIG. 9. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

TABLE 6. DEGREES OF PERSPIRATION FOR PERSONS AT WORK UNDER VARIOUS ATMOSPHERIC CONDITIONS

Work Rate 33,000 Ft Lb per Hour

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B
Forehead clammy	**59.0**	59.4	58.3	**69.5**	80.5	56.5
Body clammy	**50.0**	50.2	49.3	**57.0**	61.6	44.2
Body damp	**60.0**	60.3	59.3	**62.5**	69.6	49.5
Beads on forehead	**68.0**	68.5	67.5	**76.0**	91.0	63.4
Body wet	**69.0**	69.6	68.5	**71.0**	82.8	53.0
Perspiration on forehead runs and drips	**78.5**	79.3	78.0	**82.0**	100.5	70.2
Perspiration runs down body	**79.0**	79.8	78.5	**81.0**	99.8	69.0

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

supply, which can be obtained more conveniently from the increase in moisture content of the ventilating current.

HEAT AND MOISTURE GIVEN UP BY HUMAN BODY

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation, from the human body, which in conjunction with other heat loads (see Chapters 5 and 7) determine the capacity of the conditioner. The data in common use are those of the A.S.H.V.E. Research Laboratory[46] shown in Figs. 8, 9, 10 and 11. Other useful data are given in Tables 4, 5 and 6, which are self-explanatory.

ULTRA-VIOLET RADIATION AND IONIZATION

In spite of the rapid advances in the field of air conditioning during the past few years, the secret of reproducing indoor atmospheres of as stimulating qualities as those existing outdoors under ideal weather conditions, has not as yet been found. Extensive studies have failed to elucidate the cause of the stimulating quality of country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical means. Ultra-violet light and ionization have been suggested but the evidence so far is inconclusive or negative[47].

[46]Thermal Exchanges between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Hygiene*, Vol. XIII, No. 2, March, 1931, pp. 415-431).

[47]Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 191). Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S. H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 357). Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 271). The Nature of Ions in Air and Their Possible Physiological Effects, by L. B. Loeb (A.S. H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 101). The Influence of Ionized Air upon Normal Subjects, by L. P. Herrington (*Journal Clinical Investigation*, 14, January, 1935). The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L. P. Herrington and Karl L. Smith (*Journal Ind. Hygiene*, 17, November, 1935). Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C.-E. A. Winslow and L. P. Herrington (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 119).

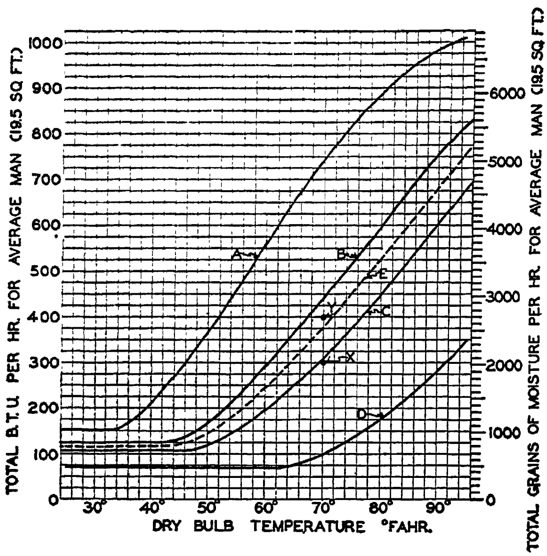

FIG. 10. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures.

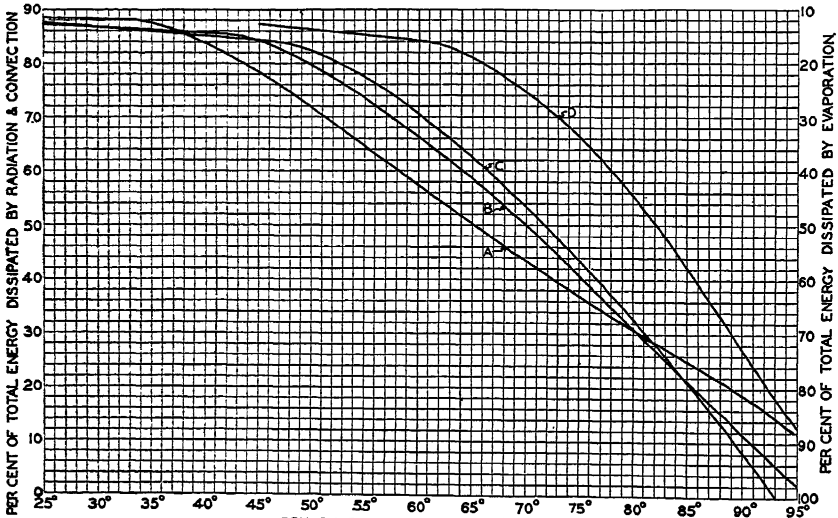

FIG. 11. HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures.

NATURAL AND MECHANICAL VENTILATION

Under favorable conditions natural ventilation methods properly combined with means for heating may be sufficient to provide for the foregoing objectives in homes, uncrowded offices, small stores, etc.

In large offices, large school rooms, and in public and industrial buildings, natural ventilation is uncertain and makes heating difficult. The chief disadvantage of natural methods is the lack of control; they depend largely on weather and upon the velocity and direction of the wind. Rooms on the windward side of a building may be difficult to heat and ventilate on account of drafts, while rooms on the leeward side may not receive an adequate amount of air from out-of-doors. The partial vacuum produced on the leeward side under the action of the wind may even reverse the flow of air so that the leeward half of the building has to take the *drift* of the air from the rooms of the windward half. Under such conditions no outdoor air would enter through a leeward window opening, but room air would pass out.

In warm weather natural methods of ventilation afford little or no control of indoor temperature and humidity. Outdoor smoke, dust and noise constitute other limitations of natural methods.

RECIRCULATION AND OZONE

The amount of recirculated air may be varied to suit changes in weather and seasonal requirements, so as to conserve heat in winter and refrigeration in summer, but the saving in operating cost should not be obtained at the expense of air quality.

Ozone has been used for deodorizing recirculated air by oxidation or *masking*. Under favorable conditions some success is possible but from the practical standpoint it is difficult to regulate the ozone output so as to just neutralize undesirable odors at all times during the occupancy of a room. The difficulties appear to be mainly due to a wide variability in the rate of ozone disappearance in different rooms, or in the same room at different times, according to the characteristics of a room, the absolute humidity, impurities in the air, number and type of occupants, and probably other factors which require considerable study before ozone can be safely and economically applied.

The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 parts of O_3 per million parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort in man, headaches and depression, lowers the metabolism, and may even lead to coma[48].

Toilets, kitchens, and similar rooms, in buildings using recirculation, should be ventilated separately by mechanical exhaust in order to prevent objectionable odors from diffusing into other parts of the building.

[48]*The British Medical Journal.* Editorial, June 25, 1932, p. 1182. See also Loc. Cit. Note 5.

PROBLEMS IN PRACTICE

1 ● What are the most comfortable air conditions?

Comfort standards are not absolute, but they are greatly affected by the physical condition of the individual, and the climate, season, age, sex, clothing, and physical activity. For the northeastern climate of the United States, the conditions which meet the requirements of the majority of people consist of temperatures between 68 and 72 F in winter and between 70 and 85 F in summer, the latter depending largely upon the prevailing outdoor temperature. The most desirable relative humidity range seems to be between 30 and 60 per cent.

2 ● Are the optimum conditions for comfort identical with those for health?

There are no absolute criteria of the prolonged effects of various air conditions on health. For the present it can be only inferred that bodily discomfort may be an indication of adverse conditions leading to poor health.

3 ● Given dry-bulb and wet-bulb temperatures of 76 F and 62 F, respectively, and an air velocity of 100 fpm, determine: (1) effective temperature of the condition; (2) effective temperature with calm air; (3) cooling produced by the movement of the air.

(1) In Fig. 1 draw line *AB* through given dry- and wet-bulb temperatures. Its intersection with the 100 ft velocity curve gives 69 deg for the effective temperature of the condition. (2) Follow line *AB* to the right to its intersection with the 20 fpm velocity line, and read 70.4 deg for the effective temperature for this velocity or so-called still air. (3) The cooling produced by the movement of the air is $70.4 - 69 = 1.4$ deg ET.

4 ● Assume that the design of an air conditioning system for a theater is to be based on an outdoor dry-bulb temperature of 95 F and a wet-bulb temperature of 78 F with an indoor relative humidity of 50 per cent. According to Table 2, the dry-bulb temperature in the auditorium should be 80 F. Estimate the sensible and latent heat given up per person.

The sensible heat given up per person per hour may be obtained from Fig. 9. With an abscissa value of 80 F, Curve *D* for men seated at rest gives a value (on the ordinate scale) of 220 Btu per person per hour as the sensible heat loss. The latent heat given up by a person seated at rest may be obtained from Fig. 10. With an abscissa value of 80 F, Curve *D* indicates a latent heat loss of 175 Btu per hour (left hand scale) or a moisture loss of 1190 grains per hour (right hand scale).

5 ● Neglecting the gain or loss of heat by transmission or infiltration through walls, windows and doors, how many cubic feet of outside air, with dry- and wet-bulb temperatures of 65 F and 59 F, respectively, (63.1 deg ET) must be supplied per hour to an auditorium containing 1000 people in order that the inside temperature shall not exceed 75 F dry-bulb and 65 F wet-bulb?

Figs. 9 and 10 give 265 Btu sensible heat and 905 grains of moisture per person with a dry-bulb temperature of 75 F in the auditorium. Therefore, 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be absorbed in raising the dry-bulb temperature from 65 to 75 F, and $265,000 \div 2.4 = 110,400$ lb of air or $110,400 \times 13.4 = 1,479,000$ cfh of air will be required. This is equivalent to $1,479,000 \div (1000 \times 60) = 24.7$ cfm per person.

The moisture content of the inside air is 76 grains per pound of dry air and that of the outside condition is 65 grains. From a psychrometric chart the increase in moisture content will therefore be 11 grains per pound of dry air. Hence $905,000 \div 11.0 = 82,300$ lb of air at the specified condition will be required. This is equivalent to $82,300 \times 13.4 = 1,103,000$ cfh of air or $1,103,000 \div (1000 \times 60) = 18.4$ cfm of air per person.

The higher volume of 24.7 cfm per person will be required to keep the dry-bulb temperature from rising above the 75 F specified. The wet-bulb temperature will therefore not rise to the maximum of 65 F.

Chapter 4

AIR POLLUTION

Classification of Air Impurities, Dust Concentrations, Air Pollution and Health, Occlusion of Solar Radiation, Smoke and Air Pollution Abatement, Dust and Cinders, Nature's Dust Catcher

THE particulate impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silks, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon.

CLASSIFICATION OF AIR IMPURITIES

The most conspicuous sources of atmospheric pollution may be classified arbitrarily according to the size of the particles as dusts, fumes, and smoke. *Dusts* consist of particles of solid matter varying from 1.0 to 150 microns in size. (micron = 0.001 millimeter or 1/25,000 in.) *Fumes* include particles resulting from chemical processing, combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size. The word fumes may be applied also to mixtures of mists (liquid droplets) and gases as *acid mists*. *Smoke* is composed of fine soot or carbon particles, usually less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health.

The lines of demarcation in these three classifications are neither sharp nor positive, but the distinction is descriptive of the nature and origin of of the particles, and their physical action. Dusts settle without appreciable agglomeration, fumes tend to aggregate, smoke to diffuse. Particles which approach the common bacteria in size—about 1 micron—are difficult to remove from air and are apt to remain in suspension unless they can be agglomerated by artificial means. The term *fly-ash* is usually applied to the microscopic glassy spheres which form the principal solid constituent of the effluent gases from powdered-coal fired furnaces. *Cinders* denote the larger solid constituents which may be entrained by furnace gases.

It is well established that particles larger than about 1 micron are unlikely to remain suspended in air currents of moderate strength. Only

violent air motion will sustain them in air long enough for them to be breathed. This means that, in hygenic problems, the engineer is concerned mostly with suspensions of particles comparable to the common

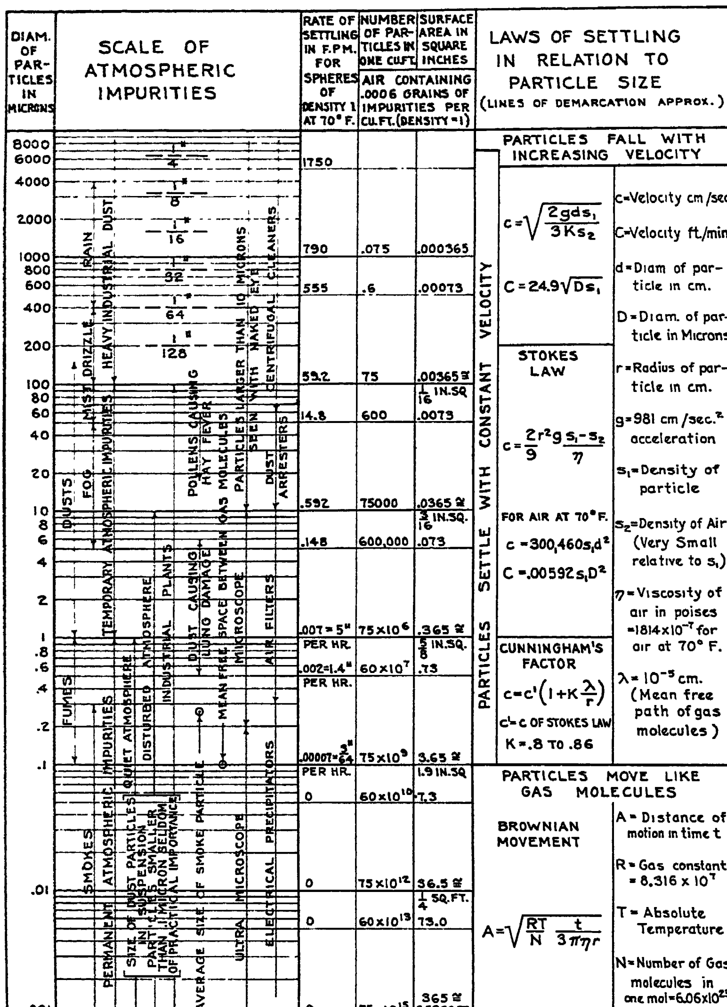

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FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

bacteria in size. A notable exception to this size limitation is the common hay-fever producing pollen such as that from rag-weed. Pollen grains may be anything from fragments 15 microns in diameter to whole pollens 25 microns or more in size. Since the lower limit of visability to the

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

GAS OR VAPOR	LOWER LIMIT VOLUME IN PER CENT	HIGHER LIMIT VOLUME IN PER CENT
Hydrogen	4.1	74.0
Ammonia	16.0	27.0
Hydrogen sulphide	4.3	46.0
Carbon disulphide	1.0	50.0
Carbon monoxide	12.5	74.0
Methane	5.3	14.0
Methane (turbulent mixture)	5.0	15.0
Ethane	3.2	12.5
Propane	2.4	9.5
Butane	1.9	8.5
Pentane	1.45	7.5
Ethylene	3.0	29.0
Acetylene	3.0	....
Acetylene (turbulent mixture)	2.3	....
Benzene	1.4	7.0
Toluene	1.4	7.0
Cyclohexane	1.3	8.3
Methyl cyclohexane	1.2	....
Methyl alcohol	7.0	....
Ethyl alcohol	4.0	19.0
Ethyl ether	1.7	26.0
Benzine	1.1	....
Gasoline	1.4	6.0
Water gas	6 to 9	55 to 70
Ethylene oxide	3.0	80.0
Acetaldehyde	4.0	57.0
Furfural (125 C)	2.0	....
Acetone	3.0	11.0
Acetone (turbulent mixture)	2.5	....
Methyl ethyl ketone	2.0	12.0
Methyl formate	6.0	20.0
Ethyl formate	3.5	16.5
Methyl acetate	4.1	14.0
Ethyl acetate	2.5	11.5
Propyl acetate	2.0	....
Butyl acetate (30 C)	1.7	....
Ethyl nitrite	3.0	....
Methyl chloride	8.0	19.0
Methyl bromide	13.5	14.5
Ethyl chloride	4.0	15.0
Ethyl bromide	7.0	11.0
Ethylene dichloride	6.0	16.0
Dichlorethylene	10.0	13.0
Vinyl chloride	4.0	22.0
Pyridine (70 C)	1.8	12.5
Natural gas	4.8	13.5
Illuminating gas	5.3	31.0
Blast-furnace gas	35.0	74.0

[a]Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones, (*U. S. Bureau of Mines, Bulletin* No. 279, 1931).

average eye is around 50 microns all air floated material of this kind is too small to identify without the aid of the microscope.

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even further. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and occasionally the engineer is confronted with the problem of removing such material before the air in question is suitable for use in building ventilation.

The physical properties of the particulate impurities of air are summarized conveniently in the chart of Fig. 1.

In the case of gases the physical property which is probably of most importance is inflammability. The best data available at present on this subject are given in Table 1.

Dust Concentrations

It is customary to report dust concentrations as grains per 1000 cu ft or milligrams per cubic meter. Gas concentrations are commonly recorded as milligrams per cubic meter or as parts per million or as per cent by volume. Typical ranges in dust concentrations as now found in practical applications are given in Table 2.

TABLE 2. DUST CONCENTRATION RANGES IN PRACTICAL APPLICATIONS[a]

Application	Grains Per 1000 Cu Ft	Mgs Per Cu M
Rural and suburban districts	0.2 to 0.4	0.4 to 0.8
Metropolitan districts	0.4 to 0.8	0.9 to 1.8
Industrial districts	0.8 to 1.5	1.8 to 3.5
Dusty factories or mines	4.0 to 80.0	10 to 200
Explosive concentrations (as of flour or soft coal)	4000 to 8000	10,000 to 20,000

[a]1 grain per 1000 cu ft = 2.3 mgs per cubic meter; 1 oz per cubic foot = 1 gram per liter.

The engineer frequently desires information regarding the effects of various concentrations of gases or dusts upon man, as the success of a particular installation may depend upon the maintenance of air which is adequately clean. At the present time there are a number of organizations working on this problem all of them publishing literature of various kinds.[1] References to books covering the hygienic significance, determination and control of dust are listed at the end of this chapter.

[1]*National Institute for Health*, U. S. Public Health Service; Division of Labor Standards, U. S. Department of Labor; University of Toronto Medical School, Canada; Saranac Laboratories, Saranac Lake, N. Y.; Air Hygiene Foundation, Inc., Pittsburgh, Pa.; Harvard School of Public Health, Boston, Mass.; Haskell Laboratory, Wilmington, Del.; and the Departments of Health and of Labor in the United States and in various provinces of Canada.

AIR POLLUTION AND HEALTH

The prevention of various diseases which result from exposure to atmospheric impurities is an engineering problem. It is important for the engineer to insure, by proper ventilation, suitable environments for working or for general living. If the equipment used is to be successful, it must operate automatically as in the modern air conditioned theatre or railroad train.

In Table 3 are given data on permissable concentrations of various substances, gases and dusts, which occur in industry. The prudent

TABLE 3. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR[a]

VAPOR OR GAS	RAPIDLY FATAL	MAXIMUM CONCENTRATION FOR FROM ½ TO 1 HOUR	MAXIMUM CONCENTRATION FOR 1 HOUR	MAXIMUM ALLOWABLE FOR PROLONGED EXPOSURE
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	....	....	....
Hydrocyanic acid	30	1½	½	⅕
Ammonia	50–100	25	3	1
Hydrochloric acid gas	10–20	½	....	1/10
Chlorine	10	½	....	1/100
Hydrofluoric acid gas	2	1/10	....	1/33
Sulphur dioxide	4–5	½–1	....	1/10
Hydrogen sulphide	10–30	5–7	2–3	1
Carbon bisulphide	....	11	5	½
Phosphene	20	4–6	1–2	....
Arsine	2½	½	½	....
Phosgene	Over ¼	¼	....	1/100
Nitrous fumes	2½–7½	1–1½	....	⅓
Benzene	190	....	31–47	1½–3
Toluene and xylene	190	....	31–47	....
Aniline	....	....	1–1½	1/10
Nitrobenzene	....	....	1/100	1/500
Carbon tetrachloride	480	240	40	1
Chloroform	250	140	50	2
Tetrachlorethane	73	....	....	1/10
Trichlorethylene	370	....	....	....
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead dust	....	....	....	0.15 mg/cu m
Quartz dust	....	....	....	1 mg/cu m

[a]Adapted from Y. Henderson and H. Haggard. (See *Noxious Gases, 1927*, and *Lessons Learned from Industrial Gases and Fumes, Institute of Chemistry of Great Britain and Ireland*, London, 1930.)

engineer will design equipment using these bench marks as the upper limits of pollution. In general it is good practice to avoid recirculation of air which contains originally toxic substances. Obviously there may be exceptions to this rule, but it is one which is generally being followed in current practice.

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it may aggravate the disease once it has started.[2]

[2]Physiological Response of the Peritoneal Tissue to Dusts Introduced as Foreign Bodies, by Miller and Sayers (*U. S. Public Health Reports*, 49:80, 1934).

The sulphurous fumes and tarry matter in smoke are more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages, leading to asthmatic breathing and bronchitis and, in extreme cases, to death. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough *CO* to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[3] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[4] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

Recent studies[5] in Pittsburgh indicate that heavy smoke pollution is definitely unhealthful. Heretofore adequate proofs on this point were lacking.

The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution are justified.

SMOKE AND AIR POLLUTION ABATEMENT

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at

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[3]Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).

[4]Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin* No. 197, 1930).

[5]Pneumoconiosis in the Pittsburgh district, Based on a Study of 2,500 Post Mortem Examinations made in Pittsburgh Hospitals, by Schnurer et al (*Journal Industrial Hygiene*, 17:294, March, 1935).

a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 9.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 44), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious

gases, such as sulphur dioxide and sulphuric acid mist, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in Chapter 26.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coal when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

NATURE'S DUST CATCHER

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. In fact, without dust in the air to form the nuclei for rain drops it would never rain, and the earth would be continually enveloped in a cloud of vapor. However, it was found in recent studies[6] that rain was not a good air cleaner of the material below about 0.7 micron.

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[6]Atmospheric Pollution of American Cities for the years 1931-1933, by J. E. Ives et al (*U. S. Public Health Bulletin* No. 224, March, 1936).

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PROBLEMS IN PRACTICE

1 ● Classify the detrimental aspects of air pollution as it affects large industrial communities.

Air pollution may be classified (*a*) medical, as it affects the physiological functions of people; (*b*) botanical, as it affects vegetation, trees, plants, shrubs and flowers; and (*c*) physical, as it affects the discoloration and deterioration of buildings, and the nuisance of soiled interior furnishings, clothes, merchandise, etc.

2 ● Distinguish between dusts, fumes, and smokes.

Solid particles ranging in size from 1.0 micron to 150 microns are called *dusts* (micron = $\frac{1}{25,000}$ in.).

Particles resulting from sundry chemical reactions and ranging from 0.1 to 1.0 micron in size are called *fumes*.

Carbon particles less than 0.1 micron in size which generally arise from the incomplete combustion of such materials as coal, oil, or tobacco are called *smokes*.

3 ● What are some of the more important physical properties of these various groups of foreign bodies which are of importance in ventilation?

In slowly moving air, dusts tend to settle out by gravity without agglomerating to form larger particles; fumes have the tendency to form larger particles which will settle when they attain the size of approximately 1.0 micron; while smokes tend to diffuse and remain in the air as permanent impurities.

4 ● Why is atmospheric pollution an important engineering problem?

a. Certain impurities, when present in too great concentrations, cause ill health or even death.

b. High concentrations of solids occlude solar radiations.

c. Some materials cause permanent injury to parts of buildings, as sulphur fumes corrode exposed metal.

d. Extra cleaning expense is incurred in dusty localities.

e. Internal combustion engines are damaged by abrasive dusts.

5 ● How may the hazards of dust-producing industrial operations best be curtailed?

By providing mechanical exhaust ventilation sufficient to keep dust concentration at a safe level (see Table 3) and then removing foreign bodies to reduce the pollution of outside air.

6 ● How may the pollution of the atmosphere be lessened?

By compelling industrial plants to install dust catching and smoke controlling devices. In many cities the domestic heating plant is one of the most serious offenders, but these plants are too small to justify the installation of dust catchers. Public education in improved firing methods would be of considerable help in this field.

7 ● What size particles are detrimental to health?

While fairly large particles may enter the upper air passages, those found in the lungs are seldom more than 10 microns in size, and comparatively few of them are more than 5 microns. It is agreed that particles between ½ and 2 microns may be harmful; some authorities place the upper limit at about 5 microns, and some incline to extend the lower limit to 0.1 of a micron.

8 ● Is the shape of the particle of any significance?

Hard particles with sharp corners or edges have a cutting effect on the delicate mucous membranes of the upper respiratory tract which may lower the resistance of the nose and throat to acute infections. This is aggravated by the irritating effects of some chemical compounds which may be taken in with the air and which act to reduce resistance.

9 ● What are the principal meteorological effects of smoke and dust?

a. The reduction in the amount of light received. Measurements have shown that visible light may be as much as 50 per cent less intense in a smoky section of a city than in a section that is free from smoke. Ultra-violet light is reduced as much or more, and in some cases is cut out entirely for a time.

b. Smoke and dust aid in the formation and prolongation of fogs. City fogs accumulate smoke and become darker in color and very objectionable. The sun requires a longer time to disperse them, and when the water is evaporated, there is a rain of smoke and soot particles that have been entrained.

10 ● Why has not smoke abatement been more effective?

Because communities have not been made sufficiently aware of the possibilities of burning high volatile fuels smokelessly and of separating cinder and ash from the stack gases to a degree that will prevent a nuisance.

11 ● Is the abatement of dust and cinders important?

Yes. Only a small percentage of the solid emission from stacks is smoke, in the accepted popular sense; the remainder is fly-ash and cinders. While black smoke is disagreeable and its tarry matter and carbon particles soil anything with which they come in contact, the cinders and some of the ash are hard and destructive. They also, together with dusts from industrial processes, make up the irritating, air-borne solids that are breathed by individuals not working in a dusty mill or factory.

12 ● Are air-borne impurities causative factors in hay fever, bronchial asthma, and allergic disorders?

Yes. Recent medical investigations indicate that 90 per cent of seasonal hay fever and 40 per cent of bronchial asthma are caused by air-borne pollens, tree dusts, and other allergic irritants.

13 ● Name some essential requirements for the smokeless combustion of fuels.

Time, temperature, and turbulence. A study of these factors is usually of value in overcoming a smoke nuisance.

14 ● What is the Ringelmann Chart Method of comparing smoke densities?

See Chapter 44. The Ringelmann Chart consists of four cards ruled with lines having different degrees of blackness. These cards, together with a white card and a black one, are hung in a horizontal row 50 ft from the observer. At this distance the lines become invisible and the cards appear to be different shades of gray, ranging from white to black. The observer, by matching the cards against the shades of smoke coming from a stack, is able to estimate the blackness of the smoke as compared with the chart.

Chapter 5

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Methods of Heat Transfer, Coefficients, Conductivity of Homogeneous Materials, Surface Conductance Coefficients, Air Space Conductance, Practical Coefficients, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Construction, Combined Coefficients of Transmission

IN order to maintain comfortable living temperatures within a building it is necessary to supply heat at the same rate that it is lost from the building. The loss of heat occurs in two ways, by direct transmission through the various parts of the structure and by air leakage or filtration between the inside and outside of the building. The purpose of this chapter is to show methods of calculation and to give practical transmission coefficients which may be applied to various structures to determine the heat loss by direct transmission. The amount lost by air filtration is determined by different methods, as outlined in Chapter 6, and must be added to that lost by direct transmission to obtain the total heating plant requirements.

METHODS OF HEAT TRANSFER

Heat transmission between the air on the two sides of a structure takes place by three methods, namely, radiation, convection and conduction. In a simple wall built up of two layers of homogeneous materials separated to give an air space between them, heat will be received from the high temperature surface by radiation, convection and conduction. It will then bė conducted through the homogeneous interior section by conduction and carried across to the opposite surface of the air space by radiation, conduction and convection. From here it will be carried by conduction through to the outer surface and leave the outer surface by radiation, convection and conduction. The process of heat transfer through a built-up wall section is complicated in theory, but in practice it is simplified by dividing a wall into its component parts and considering the transmission through each part separately. Thus the average wall may be divided into external surfaces, homogeneous materials and interior air spaces. Practical heat transmission coefficients may be derived which will give the total heat transferred by radiation, conduction and convection through any of these component parts and if the selection and method of applying these individual coefficients is thoroughly understood it is usually a comparatively simple matter to calculate the over-all heat transmission coefficient for any combination of materials.

HEAT TRANSFER COEFFICIENTS

The symbols representing the various coefficients of heat transmission and their definitions are as follows:

U = thermal transmittance or over-all coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plaster board and hollow clay tile.

f = film or surface conductance; the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space; the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 deg F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.*:

$\frac{1}{U}$ = over-all or air-to-air resistance.

$\frac{1}{k}$ = internal resistivity.

$\frac{1}{C}$ = internal resistance.

$\frac{1}{f}$ = film or surface resistance.

$\frac{1}{a}$ = air-space resistance.

As an example in the application of these coefficients assume a wall with over-all coefficient U. Then,

$$H = A\,U\,(t - t_o) \qquad (1)$$

where

H = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases.)

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

If the heat transfer between the air and the inside surface of the wall is being considered, then,

$$H = A\, f_i\, (t - t_1) \tag{2}$$

where

f_i = inside surface conductance.

t and t_1 = the temperatures of the inside air and the inside surface of the wall respectively.

In practice it is usually the over-all heat transmission coefficient that is required. This may be determined by a test of the complete wall, or it may be obtained from the individual coefficients by calculation. The simplest method of combining the coefficients for the individual parts of the wall is to use the reciprocals of the coefficients and treat them as resistance units. The total over-all resistance of a wall is equal numerically to the sum of the resistances of the various parts, and the reciprocal of the over-all resistance is likewise the over-all heat transmission coefficient of the wall. For a wall built up of a single homogeneous material of conductivity k and x inches thick the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \tag{3}$$

If the coefficients f_i, f_o and k, together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total heat resistance.

For a compound wall built up of three homogeneous materials having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces, the total resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} \tag{4}$$

For a wall with air space construction consisting of two homogeneous materials of thicknesses x_1 and x_2 and conductivities k_1 and k_2, respectively, separated to form an air space of conductance a, the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{5}$$

Likewise any combination of homogeneous materials and air spaces can be put into the wall and the over-all resistance of the combination may be calculated by adding the resistances of the individual sections of the wall. In certain special forms of construction such as tile with irregular air spaces it is necessary to consider the conductance C of the unit as built instead of the unit conductivity k, and the resistance of the section is equal to $\frac{1}{C}$. The method of calculating the over-all heat transmission coefficient for a given wall is comparatively simple, but the selection of the proper coefficients is often complicated. In some cases the construction of the wall is such that the substituting of coefficients in the accepted formula will give erroneous results. This is the case with irregular cored

out air spaces in concrete and tile blocks, and walls in which there are parallel paths for heat flow through materials having different heat resistances. In such cases it is necessary to resort to test methods to check the calculations, and in practically all cases it has been necessary to determine fundamental coefficients by test methods.

Conductivity of Homogeneous Materials

The thermal conductivity of homogeneous materials is affected by several factors. Among these are the density of the material, the amount of moisture present, the mean temperature at which the coefficient is determined, and for fiberous materials the arrangement of fiber in the material. There are many fiberous materials used in building construction and considered as homogeneous for the purpose of calculation, whereas they are not really homogeneous but are merely considered so as a matter of convenience. In general, the thermal conductivity of a material increases directly with the density of the material, increases with the amount of moisture present, and increases with the mean temperature at which the coefficient is determined. The rate of increase for these various factors is not the same for all materials, and in assigning proper coefficients one should make certain that they apply for the conditions under which the material is to be used in a wall. Failure to do this may result in serious errors in the final coefficients.

Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be effected by any factor which has an influence on any one of these three methods of transfer. The amount of heat by radiation is controlled by the character of the surface and the temperature difference between it and the surrounding objects. The amount of heat by conduction and convection is controlled largely by the roughness of the surface, by the air movement over the surface and by the temperature difference between the air and the surface. Because of these variables the surface coefficients may be subject to wide fluctuations for different materials and different conditions. The inside and outside coefficients f_i and f_o are in general affected to the same extent by these various factors and test coefficients determined for inside surfaces will apply equally well to outside surfaces under like conditions. Values for f_i in still and moving air at different mean temperatures have been determined for various building materials at the University of Minnesota under a cooperative agreement with the Society.[1]

The relation obtained between surface conductances for different materials at mean temperatures of 20 F is shown in Fig. 1. These values were obtained with air flow parallel to the surface and from other tests in which the angle of incident between the direction of air flow and the surface was varied from zero to 90 deg it would appear that these values might be lowered approximately 15 per cent for average conditions. While for average building materials there is a difference due to mean temperature, the greatest variation in these coefficients is caused by the character of the surface and the wind velocity. If other surfaces, such as

[1]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429).

aluminum foil with low emissivity coefficients were substituted, a large part of the radiant heat would be eliminated. This would reduce the total coefficient for all wind velocities by about 0.7 Btu and would make but very little difference for the higher wind velocities. In many cases in building construction the heat resistance of the internal parts of the wall is high as compared with the surface resistance and the surface factors become of small importance. In other cases such as single glass windows the surface resistances constitute practically the entire resistance of the structure, and therefore become important factors. Due to the wide variation in surface coefficients for different conditions their selection for

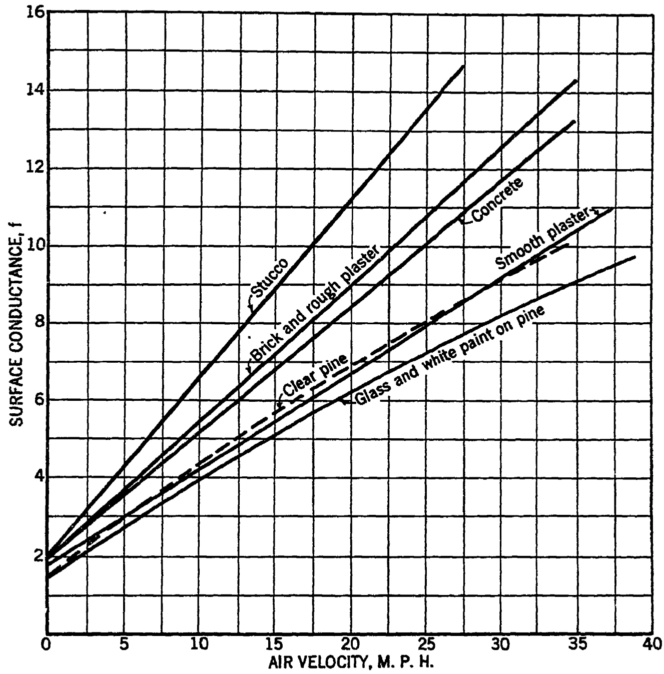


FIG. 1. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

a practical building becomes a matter of judgment. In calculating the over-all coefficients for the walls of Tables 3 to 12, 1.65 has been selected as an average inside coefficient and 6.0 as an average outside coefficient for a 15-mile wind velocity. In special cases where surface coefficients become important factors in the over-all rate of heat transfer more selective coefficients may be required.

Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. The amount of heat by radiation is governed largely by the nature of the surface and the temperature difference between the boundary surfaces of the air space. Conduction and con-

vection are controlled largely by the width and shape of the air space and the roughness of the boundary surfaces. The thermal resistances of air spaces bounded by extended parallel surfaces perpendicular to the direction of heat flow and at different mean temperatures have been determined for average building materials at the University of Minnesota in a cooperative research program with the Society.

The values given in Table 1 show the results of this study and apply to air spaces bounded by such materials as paper, wood, plaster, etc., having emissivity coefficients of from 0.9 to 0.95. The conductivity coefficients decrease with air space width until a width of about ¾ in. has been reached, after which the width has but very little effect. In these

TABLE 1. CONDUCTANCES OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

MEAN TEMP DEG FAHR	CONDUCTANCES OF AIR SPACES FOR VARIOUS WIDTHS IN INCHES						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a]Thermal Resistance of Air Spaces by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 165).

coefficients radiation is a large factor, and if surfaces with low emissivity coefficients are substituted for ordinary building materials the total amount of radiant heat will be reduced. The reduction in radiant heat caused by the low emissivity surface is independent of width of air space. Air spaces properly formed in combination with metallic surfaces such as aluminum foil, coated sheet steel, and other materials having a reflective surface, possess heat repelling characteristics. Values of air spaces lined with aluminum foil on one or both sides for widths of ⅜ in. and ¾ in. are shown in Table 2 of conductivities. A low emissivity coefficient is dependent on the permanency of the reflective surface. If a bright clean surface is covered with a thin layer of corrosive material its reflectivity is appreciably reduced[2].

In comparing the conductance coefficients for air spaces with and without bright metallic surface lining it should be noted that the reduction in heat transfer is substantially as great when one surface is lined as it is when both surfaces are lined. The reason for this is that practically 95 per cent of the total radiant heat is intercepted by one surface lining

[2]Aluminum Foil Insulation (*National Bureau of Standards Letter Circular* No. LC465, June, 1936).

and there is but a small amount left to be stopped by the second surface lining. The effect of any low emissivity surface in stopping the transmission of radiant heat is the same regardless of whether it is on the high or low temperature side of the air space. For materials such as aluminum paint or bronze paint which stop only a small percentage of radiant heat there is a greater percentage of gain by addition of a second surface lining.

PRACTICAL COEFFICIENTS

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making tests on the individual material or combination of materials. In Table 2 coefficients are given for a group of materials which have been selected from various sources. Wherever possible the properties of material and conditions of tests are given. However, in selecting and applying these values to any construction a reasonable amount of caution is necessary; variations will be found in the coefficients for the same materials, which may be partly due to different test methods used, but which are largely due to variations in materials. The recommended coefficients which have been used for the calculation of over-all coefficients as given in Tables 3 to 12 are marked by an asterisk.

It should be recognized in these tables of calculated coefficients that space limitations will not permit the inclusion of all the combinations of materials that are used in building construction and the varied applications of insulating materials to these constructions. Typical examples are given of combinations frequently used, but any special construction not given in Tables 3 to 12 can generally be computed by using the conductivity values given in Table 2 and the fundamental heat transfer formulae. For example, the tabulation of all of the values for multiple layers of insulating materials would present extensive and detailed problems of calculations for the varied application combinations, but the engineer having the fundamental conductivity values can quickly obtain the proper coefficients.

Attention is called to the fact that the conductivity values per inch of thickness do not afford a true basis for comparison between insulating materials as applied, although they are frequently used for that purpose. The value of an insulating material is measured in terms of its heat resistance, which not only depends upon the thermal conductivity coefficient per inch but also upon the thickness as installed and the manner of installation. For instance the material having a coefficient of 0.50 and 1 in. thick is equal in value to a material having a coefficient of 0.25 and a thickness of ½ in. Certain types of blanket installations are designed to be installed between the studs of a frame building in such manner as to give two air spaces. In order to get the full value of such materials they should be so installed that each air space is approximately 1 in. or more in thickness and the air spaces should be sealed at the top and bottom to prevent the circulation of air from one space to the other. Another common error in installing such a material is to nail the blanket on the outside of the studs underneath the sheathing, in which case one air space is lost and also the thickness of the insulating material is materially reduced at the studs. There are certain other types of insulation which are very porous, allowing air circulation within the material if not

properly installed. The architect or engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of good judgment in the intelligent choice of an insulating material, or its improper installation, frequently represents the difference between good or unsatisfactory results. Refer to Chapter 7 for a discussion of wall condensation.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of hard (high density) brick having a conductivity of 9.20, and that the inside course is of common (low density) brick having a conductivity of 5.0, the thicknesses each being 4 in. The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively, for still air and a 15 mph wind velocity.

Solution. k (hard high density brick) $= 9.20$; $x = 4\ 0$ in.; k (common low density brick) $= 5.0$; $x = 4.0$ in.; k (plaster) $= 3.3$; $x = ½$ in.; $f_i = 1.65$; $f_o = 6.0$. Therefore.

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

$=$ 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 3, 4 and 5 and the values of k (or C), f_i, f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example, the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.62}} = 0.48 \text{ Btu per hour per square foot per degree difference}$$

in temperature between the ground and the air immediately above the floor.

Rigid insulation refers to the so-called board form which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description: Cement	Fine Aggregate 0-No. 4	Coarse Aggregate No. 4-1½	Slump	Per Cent Voids	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
Sand and Gravel, Concrete	1	2.00	2.75	0	11.5	144.7	75.06	13.10	0.08	(4)
	1	2.75	4.50	0	10.9	145.7	74.77	12.90	0.08	(4)
	1	3.50	5.50	0	11.2	144.5	75.00	13.20	0.08	(4)
	1	2.00	2.75	5	13.9	142.5	75.50	12.10	0.08	(4)
	1	2.00	2.75	5	13.9	142.5	74.74	12.40	0.08	(4)
	1	2.75	4.50	5	14.6	141.1	73.30	12.40	0.08	(4)
	1	2.75	4.50	5	14.6	141.1	74.89	12.10	0.08	(4)
	1	3.50	5.50	5	14.7	139.2	74.50	12.85	0.08	(4)
	1	3.50	5.50	5	14.7	139.2	75.15	12.50	0.08	(4)
	Avg. Value for Sand and Gravel Concrete					142.3	......	12.62*	......	....
Limestone Concrete	1	2.00	2.75	0	16.6	135.3	74.87	11.20	0.09	(4)
	1	2.75	4.50	0	15.4	137.8	75.18	12.00	0.08	(4)
	1	3.50	5.50	0	16.3	136.4	74.75	11.50	0.09	(4)
	1	2.00	2.75	3	20.9	130.1	74.85	10.50	0.10	(4)
	1	2.75	4.50	3	23.4	126.0	74.45	10.00	0.10	(4)
	1	3.50	5.50	3	23.4	127.3	75.26	9.79	0.10	(4)
	Avg. Value for Limestone Concrete					132.15	......	10.83	......	...
Cinder Concrete	1	2.00	2.75	0	18.2	103.6	75.26	4.63	0.22	(4)
	1	2.75	4.50	0	19.9	98.7	75.71	4.30	0.23	(4)
	1	3.50	5.50	0	21.4	92.0	75.72	3.73	0.27	(4)
	1	2.00	2.75	3	22.8	101.4	74.95	4.89	0.20	(4)
	1	2.75	4.50	3	26.0	94.0	75.20	4.38	0.23	(4)
	1	3.50	5.50	3	24.4	94.4	75.55	4.24	0.24	(4)
	Avg. Value for Cinder Concrete					97.35	......	4.86	......	...
Haydite	1	2.00	2.75	0	18.0	80.7	74.82	4.15	0.25	(4)
	1	2.75	4.50	0	19.8	75.0	75.75	3.78	0.26	(4)
	1	3.50	5.50	0	21.8	71.7	74.82	3.67	0.27	(4)
	1	2.00	2.75	4	21.2	78.8	74.76	4.38	0.23	(4)
	1	2.75	4.50	4	22.2	72.4	75.39	3.89	0.26	(4)
	1	2.75	4.50	4	22.2	72.4	75.49	3.86	0.26	(4)
	1	3.50	5.50	4	23.9	71.0	75.46	4.00	0.25	(4)
	Avg. Value for Haydite					74.57	......	3.96	......	...

AUTHORITIES:

[1]U. S. Bureau of Standards, tests based on samples submitted by manufacturers.

[2]A. C. Willard, L. C. Lichty, and L. A. Harding, tests conducted at the University of Illinois.

[3]J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.

[4]F. B. Rowley, tests conducted at the University of Minnesota.

[5]A.S.H.V.E. Research Laboratory.

[6]E. A. Allcut, tests conducted at the University of Toronto.

[7]Lees and Charlton.

[8]G. B. Wilkes, tests conducted at the Massachusetts Institute of Technology.

*Recommended conductivities and conductances for computing heat transmission coefficients.

†For thickness stated or used on construction, not per 1-in. thickness.

[a]For additional conductivity data see Chapters 3 and 15, 1937 *A.S.R.E. Data Book*.

[b]If outside surface of block is painted with an impervious coat of paint, add 0.07 to resistance for sand and gravel blocks. Add 0.18 to resistance for cinder blocks. Add 0.17 to resistance for haydite blocks.

[c]Recommended value. See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[d]See A.S.H.V.E. Research Paper, Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

[e]The 6-in., 8-in., and 10-in. hollow tile figures are based on two cells in the direction of heat flow. The 12-in. hollow tile is based on three cells in the direction of heat flow. The 16-in. hollow tile consists of one 10-in. and one 6-in. tile, each having two cells in the direction of heat flow.

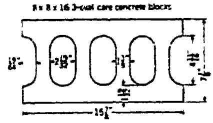
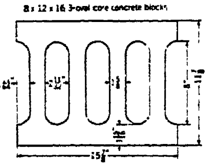
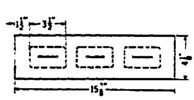
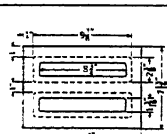
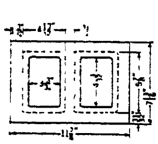
[f]Not compressed.

[g]Roofing, 0.15-in. thick (1.34 lb per sq ft), covered with gravel (0.83 lb per sq ft), combined thickness assumed 0.25.

[h]Surface values were obtained on vertical surfaces and varying with the temperature differences. A foil lined air space in a horizontal position will have a coefficient roughly three times greater if the heat flow is upward rather than downward.

TABLE 2. CONDUCTIVITIES (*k*) AND CONDUCTANCES (*C*) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description: Cement	Fine Aggregate 0-No. 4	Coarse Aggregate No. 4-½	Slump	Per Cent Voids	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (*k*) or Conductance (*C*)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
EXPANDED BURNED CLAY	1	8.00			18.4	57.9	75.57	2.28	0.44	(4)
STEAM TREATED LIMESTONE SLAG	1	7.00			27.1	74.6	74.49	2.27	0.44	(4)
PUMICE MINED IN CALIF.	1	8.00			26.5	65.0	74.68	2.42	0.41	(4)
BY-PRODUCT OF MANUFACTURE OF PHOSPHATES	1	8.00	Fineness Modulus 3.75		25.5	86.6	74.62	3.19	0.31	(4)
	1	8.00			21.1	91.1	74.43	3.42	0.29	(4)
HAYDITE	1	8.50			21.8	67.1	75.89	2.89	0.35	(4)
	1	8.50			21.8	67.1	74.60	2.815	0.34	(4)
8 x 8 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					126.4	40	0.900†	1.11	(4)
	Sand and gravel aggregate used for calculations							1.000†*	1.00	(4)
	Cores filled with 5.14 lb density cork						40	0.560†*	1.79	(4)
	Crushed limestone aggregate					134.3	40	0.856†*	1.17	(4)
	Cinder aggregate					86.2	40	0.577†	1.73	(4)
	Cinder aggregate used for calculations							0.600†*	1.66	(4)
	Cores filled with 69.7 lb density cinders						40	0.390†	2.56	(4)
	Cores filled with 5.12 lb density cork						40	0.250†*	4.00	(4)
	Cores filled with 14.2 lb density rock wool						40	0.266†*	3.76	(4)
	Haydite aggregate					67.7	40	0.495†*	2.02	(4)
	Cores filled with 5.06 lb density cork						40	0.206†	4.85	(4)
8 x 12 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					124.9	40	0.777†	1.29	(4)
	Sand and gravel aggregate used for calculations							0.800†*		(4)
	Cinder aggregate					86.2	40	0.531†*	1.88	(4)
	Cores filled with 5.24 lb density cork						40	0.237†*	4.22	(4)
	Haydite aggregate					76.7	40	0.468†	2.13	(4)
	Cores filled with 5.6 lb density cork						40	0.168†*	5.94	(4)
	Cinder aggregate					100.0	40	1.000†	1.00	(4)
	Double wall with 1 in. air space between					100.0	40	0.358†	2.70	(4)
	1 in. space filled with 9.97 lb density rock wool					100.0	40	0.204†	4.90	(4)
	5 x 8 x 12 block sand and gravel aggregate					133.7	40	0.380†	2.63	(4)
	5 x 8 x 12 block sand and gravel aggregate[b]					134.0	40	0.947†	1.06	(4)

For notes see Page 97.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness. unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS						
Brick	Low density	......	......	5.00*	0.20	....
	High density	......	......	9.20*	0.11	....
Brickwork	Damp or wet	......	......	5.00[c]	0.20	(2)
Cement Mortar	Typical	......	......	12.00*	0.08	....
Concrete	Various ages and mixes[d]	......	......	11.35 *to* 16.36	......	(5)
	Cellular	40.0	75	1.06	0.94	(3)
	Cellular	50.0	75	1.44	0.69	(3)
	Cellular	60.0	75	1.80	0.56	(3)
	Cellular	70.0	75	2.18	0.46	(3)
	Typical fiber gypsum, 87.5% gypsum and 12.5% wood chips	51.2	74	1.66*	0.60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101.0	70	3.98	0.25	(3)
Stone	Typical	......	......	12.50*	0.08	....
Stucco	Typical	......	......	12.00*	0.08	....
Tile	Typical hollow clay (4 in.)	......	......	1.00†*	1.00	....
	Typical hollow clay (6 in.)[e]	......	......	0.64†*	1.57	....
	Typical hollow clay (8 in.)[e]	......	......	0.60†*	1.67	....
	Typical hollow clay (10 in.)[e]	......	......	0.58†*	1.72	....
	Typical hollow clay (12 in.)[e]	......	......	0.40†*	2.50	....
	Typical hollow clay (16 in.)[e]	......	......	0.31†*	3.23	....
	Hollow clay (2 in.) ½-in. plaster both sides	120.0	110	1.00†	1.00	(2)
	Hollow clay (4 in.) ½-in. plaster both sides	127.0	100	0.60†	1.67	(2)
	Hollow clay (6 in.) ½-in. plaster both sides	124.3	105	0.47†	2.13	(2)
	Hollow gypsum (4 in.)	......	......	0.46†*	2.18	....
	Solid gypsum	51.8	70	1.66	0.60	(4)
	Solid gypsum	75.6	76	2.96	0.34	(4)
Tile or Terrazzo	Typical flooring	......	......	12.00*	0.08	....
INSULATION—BLANKET OR FLEXIBLE TYPES						
Fiber	Typical	......	—	0.27*	3.70	—
	Chemically treated wood fibers held between layers of strong paper[f]	3.62	70	0.25	4.00	(3)
	Eel grass between strong paper[f]	4.60	90	0.26	3.85	(1)
	" " " " "	3.40	90	0.25	4.00	(1)
	Flax fibers between strong paper[f]	4.90	90	0.28	3.57	(1)
	Chemically treated hog hair between kraft paper[f]	5.76	71	0.26	3.85	(3)
	Chemically treated hog hair between kraft paper and asbestos paper[f]	7.70	71	0.28	3.57	(3)
	Hair felt between layers of paper[f]	11.00	75	0.25	4.00	(3)
	Kapok between burlap or paper[f]	1.00	90	0.24	4.17	(1)
	Jute fiber[f]	6.70	75	0.25	4.00	(3)
	Ground paper between two layers, each ⅜-in. thick made up of two layers of kraft paper (sample ¾-in. thick)	12.1	75	0.40†	2.50	(4)
INSULATION—SEMI-RIGID TYPE						
Fiber	Felted cattle hair[f]	13.00	90	0.26	3.84	(1)
	" " "	11.00	90	0.26	3.84	(1)
	Flax[f]	12.10	70	0.30	3.33	(3)
	Flax and rye[f]	13.60	90	0.32	3.12	(1)
	Felted hair and asbestos[f]	7.80	90	0.28	3.57	(1)
	75% hair and 25% jute[f]	6.30	90	0.27	3.70	(1)
	50% hair and 50% jute[f]	6.10	90	0.26	3.85	(1)
	Jute[f]	6.70	75	0.25	4.00	(3)
	Felted jute and asbestos[f]	10.00	90	0.37	2.70	(1)
	Compressed peat moss	11.00	70	0.26	3.84	(3)
INSULATION—LOOSE FILL OR BAT TYPE						
Fiber	Made from ceiba fibers[f]	1.90	75	0.23	4.35	(3)
	" " " "	1.60	75	0.24	4.17	(3)

For notes see page 97.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—LOOSE FILL OR BAT TYPE —Continued						
Fiber	Fibrous material made from dolomite and silica	1.50	75	0.27	3.70	(3)
	Fibrous material made from slag	9.40	103	0.27	3.70	(1)
Glass Wool	Fibrous material 25 to 30 microns in diameter, made from virgin bottle glass	1.50	75	0.27	3.70	(3)
Granular	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
	Made from expanded aluminum-magnesium silicate	6.20	42	0.32	3.12	(3)
Gypsum	Cellular, dry	30.00	90	1.00	1.00	(1)
	" "	24.00	90	0.77	1.30	(1)
	" "	18.00	90	0.59	1.69	(1)
	" "	12.00	90	0.44	2.27	(1)
	Flaked, dry and fluffy[f]	34.00	90	0.60	1.67	(1)
	" " " "	26.00	90	0.52	1.92	(1)
	" " " "	24.00	75	0.48*	2.08	(3)
	" " " "	19.80	90	0.35	2.86	(1)
	" " " "	18.00	75	0.34	2.94	(3)
Mineral Wool	All forms, typical	—	—	0.27*	3.70	—
Regranulated Cork	About 3/16-in. particles	8.10	90	0.31	3.22	(1)
Rock Wool	Fibrous material made from rock	21.00	90	0.30	3.33	(1)
	" " " " "	18.00	90	0.29	3.45	(1)
	" " " " "	14.00	90	0.28	3.57	(1)
	" " " " "	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(1)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
Sawdust	Various	12.00	90	0.41	2.44	(1)
Shavings	Various from planer	8.80	90	0.41	2.44	(1)
	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
INSULATION—RIGID						
Corkboard	Typical	—	—	0.30*	3.33	—
	No added binder	14.00	90	0.34	2.94	(1)
	" " "	10.60	90	0.30	3.33	(1)
	" " "	7.00	90	0.27	3.70	(1)
	" " "	5.40	90	0.25	4.00	(1)
	Asphaltic binder	14.50	90	0.32	3.12	(1)
Fiber	Typical	—	—	0.33*	3.03	—
	Chemically treated hog hair covered with film of asphalt	10.00	75	0.28	3.57	(3)
	Made from corn stalks	15.00	71	0.33	3.03	(3)
	" " exploded wood fibers	17.90	78	0.36	3.12	(4)
	" " hard wood fibers	15.20	70	0.32	3.12	(3)
	Insulating plaster 9/10-in. thick applied to ⅜-in. plaster board base	54.00	75	1.07†	0.93	(3)
	Made from licorice roots	16.10	81	0.34	2.94	(3)
	Made from 85% magnesia and 15% asbestos	19.30	86	0.51	1.96	(1)
	Made from shredded wood and cement	24.20	72	0.46	2.17	(3)
	" " sugar cane fiber	13.50	70	0.33	3.03	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.80	70	0.30	3.33	(3)
	Made from wheat straw	17.00	68	0.33	3.03	(3)
	" " wood fiber	15.90	72	0.33	3.03	(3)
	" " " "	15.00	70	0.33	3.03	(3)
	" " " "	—	52	0.33	3.03	(6)
	" " " "	8.50	72	0.29	3.45	(3)
	" " " "	15.20	—	0.33	3.03	(3)
	" " " "	16.90	90	0.34	2.94	(1)
BUILDING BOARDS						
Asbestos	Compressed cement and asbestos sheets	123.00	86	2.70	0.37	(1)
	Corrugated asbestos board	20.40	110	0.48	2.08	(2)

For notes see Page 97.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS —Continued						
Asbestos	Pressed asbestos mill board	60.50	86	0.84	1.19	(1)
	Sheet asbestos	48.30	110	0.29	3.45	(2)
Gypsum	Gypsum between layers of heavy paper	62.80	70	1.41	0.71	(3)
	Rigid, gypsum between layers of heavy paper (½-in. thick)	53.50	90	2.60†	0.38	(1)
	Gypsum mixed with sawdust between layers of heavy paper (0.39-in. thick)	60.70	90	3.60†	0.28	(1)
Plaster Board	(⅜ in.)	—	—	3.73†*	0.27	—
	(½ in.)	—	—	2.82†*	0.35	—
ROOFING CONSTRUCTION						
Roofing	Asphalt, composition or prepared	70.00	75	6.50†*	0.15	(3)
	Built up—⅜-in. thick	—	—	3.53†*	0.28	—
	Built up, bitumen and felt, gravel or slag surfaced[g]	—	—	1.33	0.75	(2)
	Plaster board, gypsum fiber concrete and 3-ply roof covering 2½ in. thick	52.40	76	0.58†	1.72	(4)
Shingles	Asbestos	65.00	75	6.00†*	0.17	(3)
	Asphalt	70.00	75	6.50†*	0.15	(3)
	Slate	201.00	—	10.37*	0.10	(7)
	Wood	—	—	1.28†*	0.78	—
PLASTERING MATERIALS						
Plaster	Cement	—	—	8.00	0.13	(2)
	Gypsum, typical	—	—	3.30*	0.30	—
	Thickness ⅜ in.	—	73	8.80†	0.11	(4)
Metal Lath and Plaster	Total thickness ¾ in.	—	—	4.40†*	0.23	—
Wood Lath and Plaster	⅜-in. plaster, total thickness ¾ in.	—	70	2.50†*	0.40	(4)
BUILDING CONSTRUCTIONS						
Frame	1-in. fir sheathing and building paper	—	30	0.71†*	1.41	(4)
	1-in. fir sheathing, building paper, and yellow pine lap siding	—	20	0.50†*	2.00	(4)
	1-in. fir sheathing, building paper and stucco	—	20	0.55	1.82	(4)
	Pine lap siding and building paper—siding 4 in. wide	—	16	0.85†*	1.18	(4)
	Yellow pine lap siding	—	—	1.28†*	0.78	—
Flooring	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (¼-in.)	—	—	1.36†*	0.74	—
AIR SPACE AND SURFACE COEFFICIENTS						
Air Spaces	Over ¾-in. faced with ordinary building materials	—	40	1.10†*	0.91	(4)
Surfaces, Ordinary	Still air (f_i)	—	—	1.65†*	0.61	(4)
	15 mph—(f_o)	—	—	6.00†*	0.17	(4)
Surface, Rough Stucco	15 mph—(f_o)	—	—	9.00†*	0.11	(4)
Surface, Bright Aluminum	Still air (f_i)h	—	—	0.80†	1.25	(8)
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL						
	Air space, faced one side with bright aluminum foil, over ¾-in. wide	—	50	0.46†*	2.17	(4)
	Air space, faced one side with bright aluminum foil, ⅜-in. wide	—	50	0.62†	1.61	(4)
	Air space, faced both sides with bright aluminum foil, over ¾-in. wide	—	50	0.41†*	2.44	(4)
	Air space, faced both sides with bright aluminum foil, ⅜-in. wide	—	50	0.57†	1.75	(4)

For notes see Page 97.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL—Continued	Air space divided in two with single curtain of bright aluminum foil (both sides bright)					
	Each space over ¾-in. wide	......	50	0.23†*	4.35	(4)
	Each space ⅜-in. wide	......	50	0.31†	3.23	(4)
	Air space with multiple curtains of bright aluminum foil, bright on both sides, curtains more than ¾-in. apart, air circulation between spaces prevented:					
	2 curtains, forming 3 spaces	......	50	0.15†*	6.78	(4)
	3 curtains, forming 4 spaces	......	50	0.11†*	9.22	(4)
	4 curtains, forming 5 spaces	......	50	0.09†*	11.66	(4)
SPACES FACED WITH NON-METALLIC REFLECTIVE SURFACE	Fabric with non-metallic reflective surface (1/16 in. thick) placed in center of a 1½ in. air space	......	70	0.33†	3.03	(4)
	Core of fiber board coated two sides with non-metallic reflective surface (⅜ in. thick) placed in space having approximately ¾ in. air space on each side	23.4	70	0.27†	3.70	(3)
	Fiber board coated one side with non-metallic reflective surface (5/16 in. thick) placed in space having approximately ¾ in. air space on each side	......	75	0.49†	2.04	(3)
	Air space divided in two with fabric faced both sides with non-metallic reflective surface, each space over ¾-in. wide	......	40	0.33†	3.03	(4)
	Air space over ¾-in. wide faced one side with non-metallic reflective surface	......	40	0.67†	1.49	(4)
WOODS (Across Grain)						
BALSA		20.0	90	0.58	1.72	(1)
		8.8	90	0.38	2.63	(1)
		7.3	90	0.33	3.03	(1)
CALIFORNIA REDWOOD	0% moisture	22.0	75	0.66	1.53	(4)
	0% "	28.0	75	0.70	1.43	(4)
	8% "	22.0	75	0.70	1.43	(4)
	8% "	28.0	75	0.75	1.33	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	28.0	75	0.80	1.25	(4)
CYPRESS		28.7	86	0.67	1.49	(1)
DOUGLAS FIR	0% moisture	26.0	75	0.61	1.64	(4)
	0% "	34.0	75	0.67	1.49	(4)
	8% "	26.0	75	0.66	1.52	(4)
	8% "	34.0	75	0.75	1.33	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
EASTERN HEMLOCK	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	8% "	22.0	75	0.63	1.59	(4)
	8% "	30.0	75	0.81	1.23	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)
HARD MAPLE	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	8% "	40.0	75	1.08	0.93	(4)
	8% "	46.0	75	1.13	0.89	(4)
	16% "	40.0	75	1.15	0.87	(4)
	16% "	46.0	75	1.21	0.83	(4)

For notes see Page 97.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

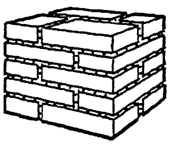
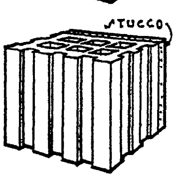
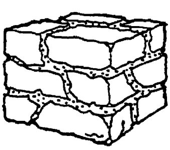
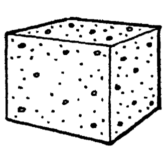
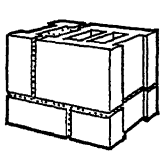
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
WOODS—Continued						
Longleaf Yellow Pine	0% moisture	30.0	75	0.76	1.32	(4)
	0% "	40.0	75	0.86	1.16	(4)
	8% "	30.0	75	0.83	1.21	(4)
	8% "	40.0	75	0.95	1.05	(4)
	16% "	30.0	75	0.89	1.12	(4)
	16% "	40.0	75	1.03	0.97	(4)
Mahogany		34.3	86	0.90	1.11	(1)
Maple		44.3	86	1.10	0.91	(1)
Maple or Oak		—	—	1.15*	0.87	—
Norway Pine	0% moisture	22.0	75	0.62	1.61	(4)
	0% "	32.0	75	0.74	1.35	(4)
	8% "	22.0	75	0.68	1.47	(4)
	8% "	32.0	75	0.83	1.21	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.91	1.10	(4)
Red Cypress	0% moisture	22.0	75	0.67	1.49	(4)
	0% "	32.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.71	1.41	(4)
	8% "	32.0	75	0.84	1.19	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.90	1.11	(4)
Red Oak	0% moisture	38.0	75	0.98	1.02	(4)
	0% "	48.0	75	1.18	0.85	(4)
	8% "	38.0	75	1.03	0.97	(4)
	8% "	48.0	75	1.24	0.81	(4)
	16% "	38.0	75	1.07	0.94	(4)
	16% "	48.0	75	1.29	0.78	(4)
Shortleaf Yellow Pine	0% moisture	26.0	75	0.74	1.35	(4)
	0% "	36.0	75	0.91	1.10	(4)
	8% "	26.0	75	0.79	1.27	(4)
	8% "	36.0	75	0.97	1.03	(4)
	16% "	26.0	75	0.84	1.19	(4)
	16% "	36.0	75	1.04	0.96	(4)
Soft Elm	0% moisture	28.0	75	0.73	1.37	(4)
	0% "	34.0	75	0.88	1.14	(4)
	8% "	28.0	75	0.77	1.30	(4)
	8% "	34.0	75	0.93	1.08	(4)
	16% "	28.0	75	0.81	1.24	(4)
	16% "	34.0	75	0.97	1.03	(4)
Soft Maple	0% moisture	36.0	75	0.89	1.12	(4)
	0% "	42.0	75	0.95	1.05	(4)
	8% "	36.0	75	0.96	1.04	(4)
	8% "	42.0	75	1.02	0.98	(4)
	16% "	36.0	75	1.01	0.99	(4)
	16% "	42.0	75	1.09	0.92	(4)
Sugar Pine	0% moisture	22.0	75	0.54	1.85	(4)
	0% "	28.0	75	0.64	1.56	(4)
	8% "	22.0	75	0.59	1.70	(4)
	8% "	28.0	75	0.71	1.41	(4)
	16% "	22.0	75	0.65	1.54	(4)
	16% "	28.0	75	0.78	1.28	(4)
Virginia Pine		34.3	86	0.96	1.04	(1)
West Coast Hemlock	0% moisture	22.0	75	0.68	1.47	(4)
	0% "	30.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.73	1.37	(4)
	8% "	30.0	75	0.85	1.18	(4)
	16% "	22.0	75	0.78	1.28	(4)
	16% "	30.0	75	0.91	1.10	(4)
White Pine		31.2	86	0.78	1.28	(1)
Yellow Pine		—	—	1.00	1.00	(3)
Yellow Pine or Fir		—	—	0.80*	1.25	—

For notes see Page 97.

TABLE 3. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL	THICKNESS OF MASONRY (INCHES)	WALL No.
	**Solid Brick** Based on 4-in. hard brick and the remainder common brick.	8 12 16	1 2 3
	**Hollow Tile** Stucco Exterior Finish. The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in. tile each having two cells in the direction of heat flow.	8 10 12 16	4 5 6 7
	**Limestone or Sandstone**	8 12 16 24	8 9 10 11
	**Concrete (Monolithic)** These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.	6 10 16 20	12 13 14 15
	Cinder (Monolithic) Conductivity k = 4.36	6 10 16 20	16 17 18 19
	Haydite (Monolithic) Conductivity k = 3.96	6 10 16 20	20 21 22 23
	**Cinder Blocks** Cores filled with dry cinders, 69.7 lb per cu ft. Cores filled with granulated cork, 5.12 lb per cu ft. Cores filled with rock wool, 14.2 lb per cu ft. Based on one air cell in direction of heat flow. Cores filled with granulated cork, 5.24 lb per cu ft.	8 8 8 8 12 12	24 25 26 27 28 29
	Concrete Blocks Cores filled with granulated cork, 5.14 lb per cu ft. Based on one air cell in direction of heat flow.	8 8 12	30 31 32
	Haydite Blocks Cores filled with granulated cork, 5.06 lb per cu ft.	8 8	33 34
	Haydite Blocks Cores filled with granulated cork, 5.6 lb per cu ft.	12 12	35 36

[a]Computed from factors marked by * in Table 2.

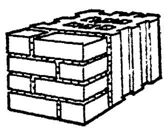
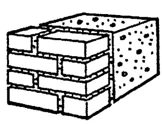
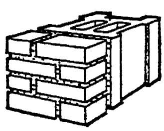
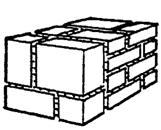
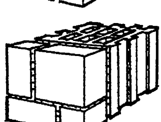
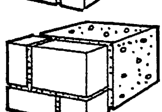
[b]Based on the actual thickness of 2-in. furring strips.

INTERIOR FINISH											
UNINSULATED WALLS						INSULATED WALLS					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	Decorated building board (½ in.) without plaster—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster (¾ in.) on metal lath attached to furring strips—furred space (over ¾-in. wide) faced one side with bright aluminum foil	Plaster on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[c]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b]) — flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.23	0.12	0.20
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.14	0.12	0.19	0.11	0.17
0.28	0.27	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.39	0.37	0.26	9.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.30	0.29	0.22	0.22	0.22	0.17	0.17	0.14	0.12	0.18	0.10	0.16
0.25	0.24	0.19	0.19	0.19	0.16	0.15	0.12	0.11	0.16	0.097	0.14
0.71	0.64	0.37	0.39	0.37	0.26	0.25	0.18	0.15	0.26	0.13	0.23
0.58	0.53	0.33	0.34	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.49	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.79	0.70	0.39	0.42	0.39	0.27	0.26	0.19	0.16	0.27	0.13	0.23
0.62	0.57	0.34	0.37	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.48	0.44	0.29	0.31	0.29	0.22	0.21	0.16	0.14	0.22	0.12	0.20
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.46	0.43	0.29	0.30	0.29	0.22	0.21	0.16	0.14	0.22	0.12	0.19
0.33	0.31	0.23	0.24	0.23	0.18	0.18	0.14	0.12	0.18	0.11	0.16
0.22	0.22	0.17	0.18	0.17	0.15	0.14	0.12	0.10	0.15	0.09	0.13
0.19	0.18	0.15	0.15	0.15	0.13	0.13	0.11	0.09	0.13	0.09	0.12
0.44	0.41	0.28	0.29	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.30	0.29	0.22	0.23	0.22	0.17	0.17	0.14	0.12	0.18	0.10	0.16
0.21	0.20	0.16	0.17	0.16	0.14	0.14	0.11	0.10	0.14	0.09	0.13
0.17	0.17	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.42	0.39	0.27	0.28	0.27	0.21	0.20	0.16	0.13	0.21	0.12	0.19
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.22	0.21	0.17	0.18	0.17	0.14	0.14	0.12	0.11	0.14	0.09	0.13
0.23	0.22	0.19	0.18	0.18	0.15	0.14	0.12	0.10	0.15	0.09	0.14
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.20	0.19	0.17	0.16	0.16	0.13	0.13	0.11	0.10	0.14	0.09	0.13
0.56	0.52	0.32	0.34	0.32	0.24	0.23	0.17	0.14	0.24	0.12	0.21
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.49	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.23	0.12	0.20
0.36	0.34	0.26	0.26	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.18	0.17	0.15	0.15	0.14	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.34	0.32	0.25	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.15	0.14	0.13	0.13	0.12	0.11	0.11	0.09	0.08	0.11	0.08	0.10

[c]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS WITH VARIOUS TYPES OF VENEERS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL — FACING	TYPE OF WALL — BACKING	WALL NO.
	4 in. **Brick Veneer**[d]	6 in. **Hollow Tile**[c]	37
		8 in. **Hollow Tile**[c]	38
		10 in. **Hollow Tile**[c]	39
		12 in. **Hollow Tile**[c]	40
	4 in. **Brick Veneer**[d]	6 in. **Concrete**	41
		10 in. **Concrete**	42
		16 in. **Concrete**	43
	4 in. **Brick Veneer**[d]	8 in. **Cinder Blocks**	44
		8 in. **Cinder Blocks** — Cores filled with granulated cork, 5.12 lb per cu ft.	45
		12 in. **Cinder Blocks**	46
		12 in. **Cinder Blocks** — Cores filled with granulated cork, 5.24 lb per cu ft.	47
		8 in. **Concrete Blocks**	48
		8 in. **Concrete Blocks**—Cores filled with granulated cork, 5.14 lb per cu ft.	49
		12 in. **Concrete Blocks**	50
		8 in. **Haydite Block**	51
		8 in. **Haydite Block**—Cores filled with granulated cork, 5.06 lb per cu ft.	52
		12 in. **Haydite Block**	53
		12 in. **Haydite Block**—Cores filled with granulated cork, 5.6 per cu ft.	54
	4 in. **Cut-Stone Veneer**[d]	8 in. **Common Brick**	55
		12 in. **Common Brick**	56
		16 in. **Common Brick**	57
	4 in. **Cut-Stone Veneer**[d]	6 in. **Hollow Tile**[c]	58
		8 in. **Hollow Tile**[c]	59
		10 in. **Hollow Tile**[c]	60
		12 in. **Hollow Tile**[c]	61
	4 in. **Cut-Stone Veneer**[d]	6 in. **Concrete**	62
		10 in. **Concrete**	63
		16 in. **Concrete**	64

[a]Computed from factors marked by * in Table 2.
[b]Based on the actual thickness of 2-in. furring strips.
[c]The 6-in., 8-in. and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile is based on three cells in the direction of heat flow.

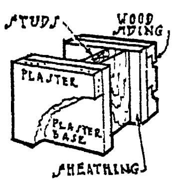
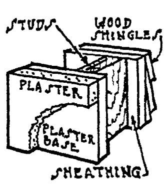
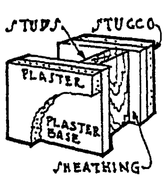
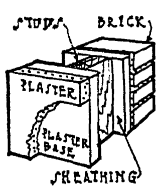
INTERIOR FINISH

Uninsulated Walls						Insulated Walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in. on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on cork board (1½ in.) set in cement mortar (½ in.)	Plaster (¾ in.) on metal lath attached to furring strips—furred space (over ¾-in. wide) faced one side with bright aluminum foil	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[f]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.16	0.13	0.19	0.11	0.17
0.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.27	0.26	0.20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
0.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0 11	0.18
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.20	0.19	0.16	0.16	0.16	0.13	0.13	0.11	0.10	0.13	0.09	0.12
0.31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.18	0.18	0.15	0.15	0.15	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.34	0.32	0.24	0.25	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.17	0.16	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.29	0.28	0.21	0.22	0.21	0.17	0.17	0.13	0.12	0.17	0.10	0.16
0.14	0.14	0.12	0.12	0.12	0.10	0.10	0.09	0.08	0.10	0.07	0.10
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
0.23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.28	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.61	0.56	0.34	0.36	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.51	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.23	0.12	0.20
0.41	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.21	0.11	0.18

[d]Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e]Based on one air cell in direction of heat flow.
[f]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FRAME CONSTRUCTION[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	EXTERIOR FINISH	TYPE OF SHEATHING	WALL No.
		1 in. Wood[d]	65
	Wood Siding or Clapboard	25/32 in. Rigid Insulation	66
		½ in. Plaster Board	67
		1 in. Wood[d]	68
	Wood Shingles	25/32 in. Rigid Insulation[e]	69
		½ in. Plaster Board[e]	70
		1 in. Wood[d]	71
	Stucco	25/32 in. Rigid Insulation	72
		½ in. Plaster Board	73
		1 in. Wood[d]	74
	Brick[f] Veneer	25/32 in. Rigid Insulation	75
		½ in. Plaster Board	76

[a]Computed from factors marked by * in Table 2.

[b]These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plaster board.

[e]Based on the actual width of 2 by 4-in. studding, namely, 3⅝ in.

INTERIOR FINISH

No Insulation Between Studding								Insulation Between Studding				
Plaster on wood lath on studding	Plaster (3/4 in.) on metal lath on studding	Plaster (1/2 in.) on plaster board (3/8 in.) on studding	Plaster (1/2 in.) on rigid insulation (1/2 in.) on studding	Plaster (1/2 in.) on rigid insulation (1 in.) on studding	Plaster (1/2 in.) on corkboard (1 1/2 in.) on studding	No plaster—decorated rigid or building board interior finish (1/2 in.)	1 in. wood sheathing,[d] furring strips, plaster (1/2 in.) on wood lath	Plaster (3/4 in.) on metal lath—stud space faced one side with bright aluminum foil	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding and in contact with sheathing	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—rock wool fill (3 5/8 in.[e]) between studding[g,h]
A	B	C	D	E	F	G	H	I	J	K	L	M
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.20	0.17	0.15	0.12	0.072
0.23	0.24	0.23	0.18	0.14	0.11	0.18	0.14	0.19	0.17	0.13	0.10	0.070
0.31	0.33	0.31	0.22	0.17	0.13	0.23	0.19	0.24	0.20	0.17	0.13	0.076
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.20	0.17	0.15	0.12	0.072
0.19	0.20	0.19	0.15	0.12	0.10	0.16	0.14	0.16	0.14	0.11	0.094	0.066
0.24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.19	0.17	0.15	0.12	0.071
0.30	0.31	0.30	0.22	0.16	0.12	0.22	0.19	0.23	0.20	0.17	0.13	0.076
0.27	0.29	0.27	0.20	0.16	0.12	0.21	0.15	0.22	0.19	0.14	0.11	0.074
0.40	0.43	0.40	0.26	0.19	0.14	0.28	0.22	0.29	0.24	0.20	0.14	0.081
0.27	0.28	0.27	0.20	0.15	0.12	0.21	0.17	0.21	0.18	0.16	0.12	0.074
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.15	0.20	0.18	0.13	0.11	0.072
0.35	0.37	0.35	0.24	0.18	0.13	0.25	0.21	0.26	0.22	0.18	0.14	0.079

[d]Yellow pine or fir—actual thickness about 25/32 in.

[e]Furring strips between wood shingles and sheathing.

[f]Small air space and mortar between building paper and brick veneer neglected.

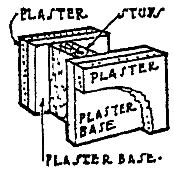
[g]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

[h]Stud and rock wool fill areas combined.

TABLE 6. COEFFICIENTS OF TRANSMISSION (*U*) OF FRAME INTERIOR WALLS AND PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



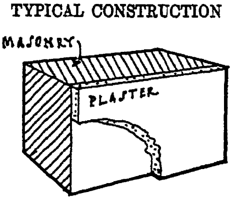
TYPICAL CONSTRUCTION	WALL No.	SINGLE PARTITION (FINISH ON ONE SIDE OF STUDDING)	DOUBLE PARTITION (FINISHED ON BOTH SIDES OF STUDDING)					
			Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½-in. Flexible Insulation Between Studding (One Air Space)	Stud Space Faced One Side with Bright Aluminum Foil
TYPE OF WALL		A	B	C	D	E	F
Wood Lath and Plaster On Studding	77	0.62	0.34	0.11	0.076	0.21	0.24
Metal Lath and Plaster[c] On Studding	78	0.69	0.39	0.11	0.078	0.23	0.26
Plaster Board (⅜ in.) and Plaster[d] On Studding	79	0.61	0.34	0.10	0.075	0.21	0.24
½ in. Rigid Insulation and Plaster[d] On Studding	80	0.35	0.18	0.083	0.063	0.14	0.15
1 in. Rigid Insulation and Plaster[d] On Studding	81	0.23	0.12	0.066	0.054	0.097	0.10
1½ in. Corkboard and Plaster[d] On Studding	82	0.16	0.081	0.052	0.044	0.070	0.073
2 in. Corkboard and Plaster[d] On Studding	83	0.12	0.063	0.045	0.038	0.057	0.059

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾-in. thick.
[d]Plaster assumed ½-in. thick.

TABLE 7. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION

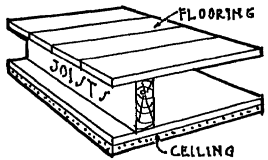


TYPE OF WALL	No.	PLAIN WALLS (NO PLASTER)	WALLS PLASTERED ON ONE SIDE	WALLS PLASTERED ON BOTH SIDES
		A	B	C
4-in. Hollow Clay Tile	84	0.45	0.42	0.40
4-in. Common Brick	85	0.50	0.46	0.43
4-in. Hollow Gypsum Tile	86	0.30	0.28	0.27
2-in. Solid Plaster	87	-------	-------	0.53

[a]Computed from factors marked by * in Table 2.

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.



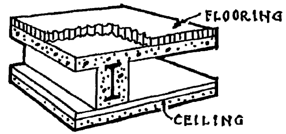
TYPICAL CONSTRUCTION	INSULATION BETWEEN JOISTS	No.	TYPE OF FLOORING				
			No Flooring	Yellow Pine Flooring[b] on Joists	Yellow Pine Flooring on Rigid Insulation (½ in.) on Joists	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring[b] on Joists	¼-in. Battleship Linoleum on Yellow Pine Flooring[b]
TYPE OF CEILING			A	B	C	D	E
No Ceiling	None	1	....	0.46	0.27	0.34	0.34
Metal Lath and Plaster (¾ in.)	None	2	0.69	0.30	0.21	0.25	0.25
Wood Lath and Plaster	None	3	0.62	0.28	0.20	0.24	0.24
Plaster Board (⅜ in.) and Plaster (½ in.)	None	4	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (½ in.) and Plaster (½ in.)	None	5	0.35	0.21	0.16	0.18	0.18
Rigid Insulation (1 in.) and Plaster (½ in.)	None	6	0.23	0.16	0.13	0.14	0.14
Metal Lath and Plaster	Bright Aluminum Foil[e]	7	0.59	0.22	0.17	0.19	0.19
Metal Lath and Plaster	Flexible[d] Insulation (1 in.)	8	0.17	0.13	0.11	0.12	0.12
Metal Lath and Plaster	Flexible[d] Insulation (2 in.)	9	0.10	0.086	0.076	0.081	0.081
Metal Lath and Plaster	Rock Wool Fill (3⅝ in.)	10	0.079	0.068	0.063	0.066	0.066
Corkboard (1½ in.) and Plaster (½ in.)	None	11	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and Plaster (½ in.)	None	12	0.12	0.10	0.087	0.094	0.094

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed to be 25/32 in.
[c]Thickness assumed to be 13/16 in.
[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in. furring strips.
[e]Air space faced on one side with bright aluminum foil.

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION

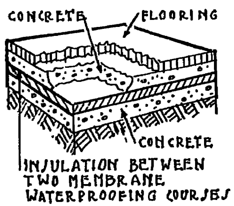


Type of Ceiling	Thickness of Concrete (Inches)	No.	No Flooring (Concrete Bare)[b]	Yellow Pine Flooring[c] on Wood Sleepers Embedded in Concrete[d]	Maple or Oak Flooring[e] on Yellow Pine Sub-Flooring[c] on Wood Sleepers Embedded in Concrete	Tile or Terrazzo[f] Flooring on Concrete	¼-in. Battleship Linoleum Directly on Concrete
			A	B	C	D	E
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44
	6	2	0.59	0.37	0.30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0.27	0.47	0.36
½ in. Plaster Applied Directly to Under Side of Concrete	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0.26	0.44	0.34
Suspended or Furred Metal Lath and Plaster (¾ in.) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0.21	0.32	0.27
	10	12	0.32	0.24	0.21	0.31	0.25
Suspended or Furred Ceiling of Plaster Board (⅜ in.) and Plaster (½ in.)	4	13	0.35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0.21	0.30	0.25
	10	16	0.30	0.23	0.20	0.29	0.24
Suspended or Furred Ceiling of Rigid Insulation (½ in.) and Plaster (½ in.)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0.18	0.16	0.21	0.19
Plaster (½ in.) on Corkboard (1½ in.) Set in Cement Mortar (½ in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

[a]Computed from factors marked by * in Table 2.
[b]The figures in COLUMN A may be used with sufficient accuracy for concrete floors covered with carpet.
[c]Thickness of yellow pine flooring assumed to be 25/32 in.
[d]The figures in COLUMN B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers.
[e]Thickness of maple or oak flooring assumed to be 13/16 in.
[f]Thickness of tile or terrazzo assumed 1 in.

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a, e]

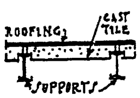
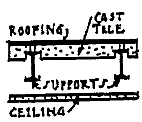
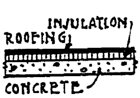
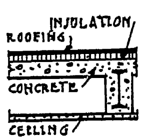
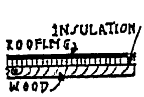
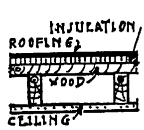
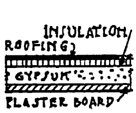
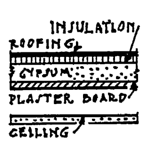
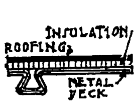
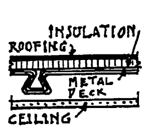
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions.



TYPICAL CONSTRUCTION	THICKNESS OF CONCRETE (INCHES)	No.	TYPE OF FINISH FLOORING				
			No Flooring (Concrete Bare)	Yellow Pine Flooring[b] on Wood Sleepers Resting on Concrete	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	Tile or Terrazzo[d] on Concrete	¼-in. Battleship Linoleum Directly on Concrete
TYPE AND THICKNESS OF INSULATION			A	B	C	D	E
None	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None[e]	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
1 in. Rigid Insulation[e]	4	7	0.22	0.16	0.14	0.22	0.19
1 in. Rigid Insulation[e]	8	8	0.21	0.15	0.13	0.20	0.18
2 in. Corkboard[e]	4	9	0.12	0.099	0.093	0.12	0.11
2 in. Corkboard[e]	8	10	0.12	0.096	0.090	0.12	0.11

[a]Computed from factors marked by * in Table 2.
[b]Assumed 25/32 in. thick.
[c]Assumed 13/16 in. thick.
[d]Assumed 1 in. thick.
[e]The figures for Nos. 5 to 10, inclusive, include 3-in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

Table 11. Coefficients of Transmission (U) of Various Types of Flat Roofs Covered with Built-Up Roofing[a]

Typical Construction: Without Ceilings	Typical Construction: With Metal Lath and Plaster Ceilings[d]	Type of Roof Deck	Thickness of Roof Deck (Inches)	No.
		**Precast Cement Tile**	1⅝	1
		**Concrete** **Concrete** **Concrete**	2 4 6	2 3 4
		**Wood** **Wood** **Wood** **Wood**	1[b] 1½[b] 2[b] 4[b]	5 6 7 8
		**Gypsum Fiber Concrete[c] (2 in.) on Plaster Board (⅜ in.)** **Gypsum Fiber Concrete[c] (3 in.) on Plaster Board (⅜ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (½ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (1 in.)**	2⅜ 3⅜ 2½ 3	9 10 11 12
		**Flat Metal Roofs** Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.	.....	13

[a]Computed from factors marked by * in Table 2.
[b]Nominal thicknesses specified—actual thicknesses used in calculations.
[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

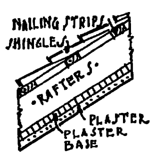
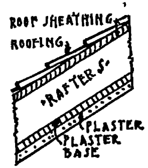
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[d]							
No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.10	0.20	0.16	0.13	0.11	0.09	0.12	0.10	0.087
0.19	0.15	0.12	0.10	0.09	0.12	0.10	0.08	0.16	0.13	0.11	0.09	0.08	0.10	0.09	0.077
0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[d]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

Typical Construction	Type of Roofing and Roof Sheathing	Insulation Between Roof Rafters	No.	Type of Ceiling (Applied Directly to Roof Rafters): A. No Ceiling (Rafters Exposed)	B. Metal Lath and Plaster (3/4 in.)	C. Plaster Board (3/8 in.) and Plaster (1/2 in.)	D. Wood Lath and Plaster	E. Rigid Insulation (1/2 in.)	F. Rigid Insulation (1/2 in.) and Plaster (1/2 in.)	G. Rigid Insulation (1 in.) and Plaster (1/2 in.)	H. Corkboard (1½ in.) and Plaster (1/2 in.)	I. Corkboard (2 in.) and Plaster (1/2 in.)
	Wood Shingles on Wood Strips[b]	None	1	0.46	0.30	0.29	0.29	0.22	0.21	0.16	0.12	0.10
		Bright Aluminum Foil[g]	2	....	0.22	0.21	0.21	0.18	0.17	0.14	0.11	0.089
		1 in. Flexible[c]	3	....	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
		2 in. Flexible[c]	4	....	0.086	0.083	0.083	0.076	0.075	0.068	0.060	0.054
		3⅝ in. Rock Wool[e]	5	....	0.063	0.062	0.062	0.058	0.058	0.053	0.048	0.044
	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing[d] on Wood Sheathing[f]	None	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
		Bright Aluminum Foil[g]	7	....	0.24	0.24	0.24	0.19	0.18	0.14	0.11	0.098
		1 in. Flexible[c]	8	....	0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
		2 in. Flexible[c]	9	....	0.088	0.087	0.087	0.079	0.078	0.070	0.062	0.056
		3⅝ in. Rock Wool[e]	10	....	0.065	0.064	0.064	0.060	0.059	0.054	0.049	0.045

[a]Computed from factors marked by * in Table 2. Nos. 6 to 10, inclusive, based on ½-in. thick slate.
[b]Based on 1 in. by 4 in. strips spaced 2 in.
[c]Figures based on two air spaces. Insulation may also be applied to under side of roof rafters with furring strips between.
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e]Assumed 3⅝ in. thick based on the actual width of 2 in. by 4 in. rafters.
[f]Sheathing assumed 25/32 in. thick.
[g]Air space faced on one side with bright aluminum foil.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall

A. Windows and Skylights

DESCRIPTION	U
Single	1.13[a, c]
Double	0.45[a]
Triple	0.281[a]

B. Solid Wood Doors[b, c]

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U
1	25/32	0.69
1¼	1 1/16	0.59
1½	1 5/16	0.52
1¾	1⅜	0.51
2	1⅝	0.46
2½	2⅛	0.38
3	2⅝	0.33

C. Glass Walls

DESCRIPTION	U
Hollow glass tile wall, 6 x 6 x 2 in. thick blocks	
Wind velocity 15 mph, outside surface; still air, inside surface	0.60
Still air, outside and inside surface	0.48

[a]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.
[b]Computed using C = 1.15 for wood; f_i = 1.65 and f_o = 6.0.
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperatures.

it is probable that the values of U for these two types of roofs will compare favorably.

The thicknesses upon which the coefficients in Tables 3 to 13, inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	¾ in.
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	½ in.
Slate (roofing)	½ in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	⅜ in.
1-in. lumber (S-2-S)	25/32 in.
1½-in. lumber (S-2-S)	1 5/16 in.
2-in. lumber (S-2-S)	1⅝ in.
2½-in. lumber (S-2-S)	2⅛ in.
3-in. lumber (S-2-S)	2⅝ in.
4-in. lumber (S-2-S)	3⅝ in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4-in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1-in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top floor ceiling, unheated attic space, and pitched roof, per square foot of ceiling area, is as follows:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \tag{6}$$

where

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

Stating the formula in terms of the total heat resistance of the ceiling and roof,

$$\frac{1}{U} = R = \frac{1}{U_{ce}} + \frac{1}{U_r \times n} \tag{7}$$

In selecting the values to be used for U_r and U_{ce} it should be noted that the under surface of the roof and the upper surface of the ceiling are more nearly equivalent to the boundary surfaces of an internal air space than they are to the external surfaces of a wall. It would be more nearly correct to use a value of 2.2 rather than the usual value of 1.65 as coefficients for these surfaces. In most cases this would make only a minor change in U. It should be noted that the over-all coefficient should be multiplied by the ceiling and not the roof area.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall spaces the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top floor ceiling. If the unheated attic contains windows and vertical wall spaces these must be taken into consideration in calculating the roof area and also its coefficient U_r. In this case an approximate value of U_r may be obtained as the summation of the coefficient of each individual section such as the roof, vertical walls or windows times its percentage of total area. This coefficient may be used with reasonable accuracy in the above formulae. If, however, there are roof ventilators such that the attic air is substantially at outside temperature, then the roof should be neglected and only the coefficient for the top floor ceiling construction used.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms

of 32 F. Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 7.

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Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 285).

Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 33).

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The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 329).

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Heat Insulation as Applied to Buildings and Structures, by E. A. Allcut, University of Toronto, 1934.

House Insulation, Its Economies and Application, by Russell E. Backstrom (Report of the National Committee on Wood Utilization, *United States Government Printing Office*, 1931).

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Calculation of Heat Transmission, by Margaret Fishenden and Owen A. Saunders.

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.

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Industrial Heat Transfer, by Shack.

PROBLEMS IN PRACTICE

1 ● What is the coefficient *U* and how is it applied?

The coefficient *U* is the heat loss through walls, ceilings, and floors and the value depends upon the construction and material, expressed in Btu per hour per square foot per degree difference in temperature between the inside and outside. To determine the total heat loss, multiply *U* for each material by the square feet of surface and the temperature difference.

2 ● Find the value of U for a 6-in. concrete wall with plaster on metal lath attached to 2-in. furring strips with flanged ½-in. blanket insulation.

0.23 (Table 3, Wall 12L).

3 ● A wall is built with two layers of ½-in. insulating material spaced 1 in. apart; the air space is lined on one side with bright aluminum foil; mean temperature is 40 F; still air on both sides of wall; *k* for insulating material is 0.34. Calculate the value of *U*.

$$f_i = 1.65; f_o = 1.65; a = 0.46$$

$$R = \frac{1}{1.65} + \frac{0.5}{0.34} + \frac{1}{0.46} + \frac{0.5}{0.34} + \frac{1}{1.65} = 6.327$$

$$U = \frac{1}{R} = 0.158$$

4 ● What is the inside surface temperature of a 6-in. solid concrete wall? Inside air, 70 F; outside air, −20 F with 15 mph wind.

The temperature drop from point to point through a wall is directly proportional to the heat resistance.

$$f_i = 1.65; k \text{ for concrete} = 12.62; f_o = 6.0$$

$$\text{Over-all resistance } R = \frac{1}{1.65} + \frac{6}{12.62} + \frac{1}{6.0} = 1.27$$

$$\frac{\text{Temperature drop, inside air to surface}}{\text{Temperature drop, air to air}} = \frac{\frac{1}{1.65}}{1.27}$$

$$\text{Temperature drop, inside air to surface} = \frac{90}{1.27 \times 1.65} = 43$$

$$70 - 43 = 27 \text{ F, inside surface temperature of wall.}$$

5 ● How many inches of insulating material having a conductivity of 0.30 would be required, for the wall of Question 4, to raise the inside surface temperature to 60 F?

Temperature drop, air to inside surface = 10 F; temperature drop, inside surface to outside air = 80 F. Therefore, the heat resistance from inside wall surface to outside air must be eight times that from inside air to inside wall surface, or $8 \times \frac{1}{1.65} = 4.85$. The resistance for added material is, therefore,

$$4.85 - \left(\frac{6}{12.62} + \frac{1}{6}\right) = 4.19$$

$$4.19 \times 0.30 = 1.25 \text{ in. of insulation.}$$

6 ● An unheated attic space in a residence has an equivalent pitched roof area of 1560 sq ft and a ceiling area of 1200 sq ft. If 15 per cent of the roof area is composed of vertical wall spaces having a value of U = 0.52, determine the total heat loss per hour through the ceiling and roof for a temperature difference of 85 F, if U = 0.46 for the roof and U = 0.38 for the ceiling.

An approximate value of *U* for the roof is equivalent to the summation of coefficients for each individual section times its percentage of total area.

$U_r = (0.52 \times 0.15) + (0.46 \times 0.85) = 0.47.$

Ratio of roof area to ceiling = 1560 ÷ 1200 = 1.3.

Substituting in Formula 6:

$$U = \frac{0.47 \times 0.38}{0.47 + \frac{0.38}{1.3}} = 0.235$$

$H = A\,U\,(t_i - t_o) = 1200 \times 0.235 \times 85 = 23{,}900$ Btu per hour.

Chapter 6

AIR LEAKAGE

Nature of Air Infiltration, Infiltration Through Walls, Window Leakage, Door Leakage, Selection of Wind Velocity, Crack Length used for Computations, Multi-Story Buildings, Heat Equivalent of Air Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 7 and 8.)

NATURE OF AIR INFILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Tests on mechanically ventilated classrooms of average construction have shown that air infiltration acts quite independently of the planned air supply. Accordingly, the heating or cooling load owing to air infiltration from natural causes should be considered in addition to the ventilating load.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-story buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating

factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

INFILTRATION THROUGH WALLS

Table 1 gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12.2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed at the end of this chapter.

[b]Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

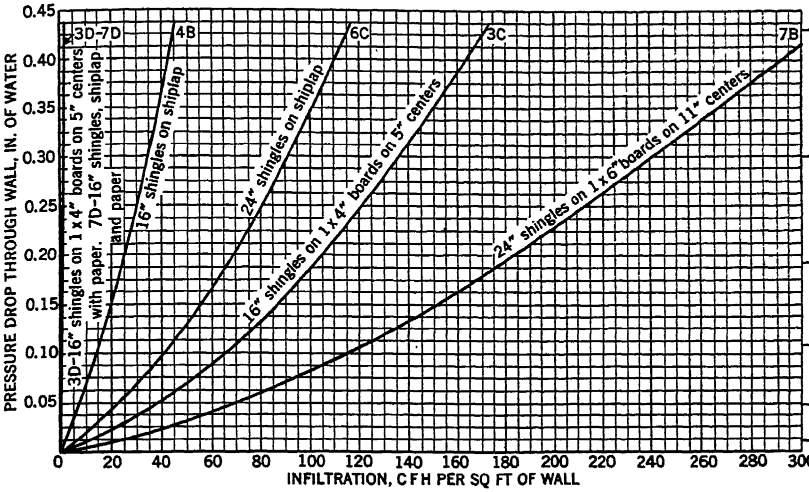

FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

The amount of infiltration that may be expected through simple walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration indicated in Figs. 1 and 2 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

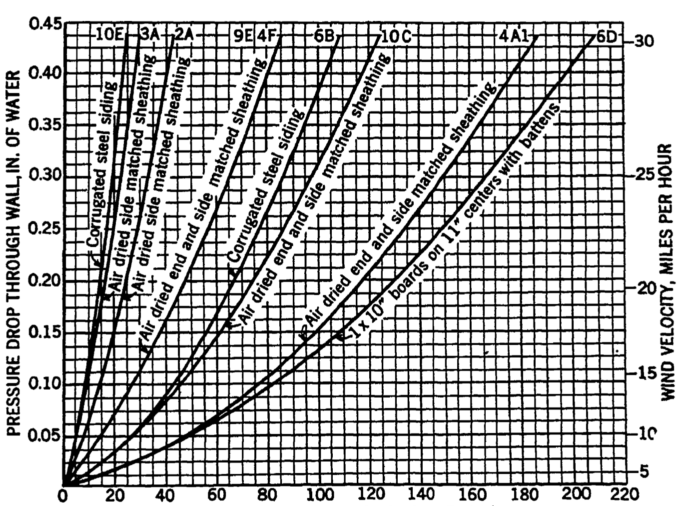

FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance as illustrated in Fig. 3. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not

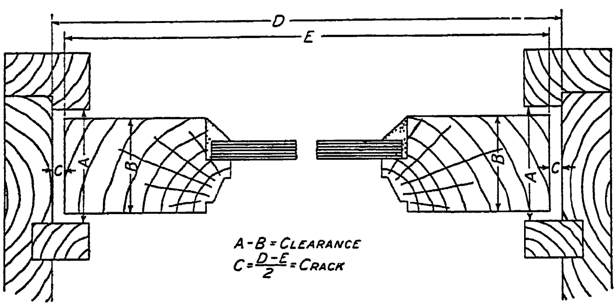

FIG. 3. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

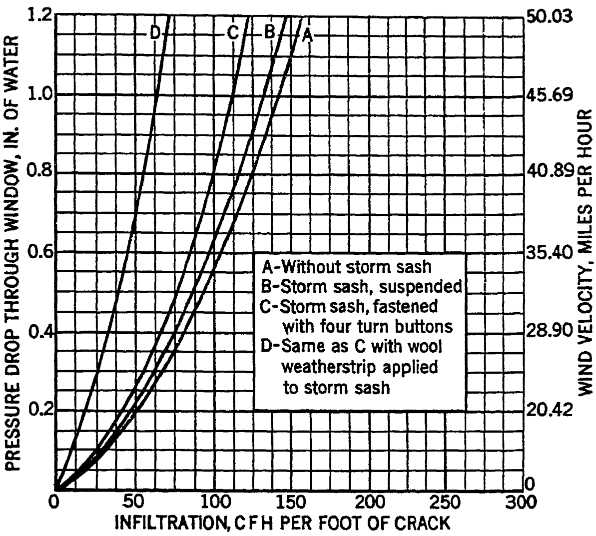

FIG. 4. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—5/64-IN. CRACK AND 1/32-IN. CLEARANCE

Leakage values for storm sash are given in Figs. 4 and 5. When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

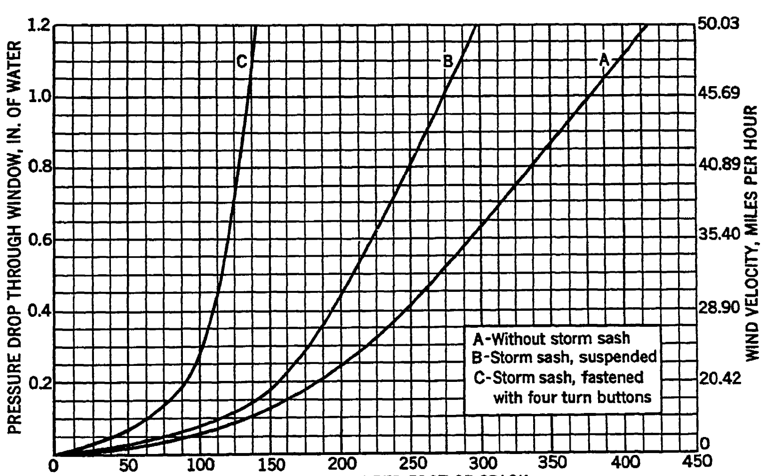

FIG. 5. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—1/8-IN. CRACK AND 1/8-IN. CLEARANCE

TABLE 2. INFILTRATION THROUGH WINDOWS

Expressed in Cubic Feet per Foot of Crack per Hour[a]

TYPE OF WINDOW	REMARKS	WIND VELOCITY, MILES PER HOUR 5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weather-stripped, 1/16-in. crack and 3/64-in. clearance[c]. Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weather-stripped, 3/32-in. crack and 3/32-in. clearance.[e] Includes wood frame leakage[d]	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted[g], 1/16-in. crack	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in. crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

DOOR LEAKAGE

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3 represent current engineering practice. These values are based on the average number of persons in a room at a specified time, which may also be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 3 and 8.

TABLE 3. INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]
Expressed in Cubic Feet per Minute per Person in Room

APPLICATION	PAIR 36 IN. SWINGING DOORS, SINGLE ENTRANCE[b]
Bank	7.5
Barber Shop	4.5
Broker's Office	7.0
Candy and Soda	6.0
Cigar Store	25.0
Department Store	8.0
Dress Shop	2.5
Drug Store	7.0
Furrier	2.5
Hospital Room	3.5
Lunch Room	5.0
Men's Shop	3.5
Office	3.0
Office Building	2.0
Public Building	2.5
Restaurant	2.5
Shoe Store	3.5

[a]For doors located in only one wall or where doors in other walls are of revolving type.

[b]Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.

Infiltration for 72 in. revolving doors may be assumed 60 per cent of swinging door values.

SELECTION OF WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Until this point is definitely established the practice of using average values will be followed. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 7.

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, as explained in Chapter 7, is to select an outside temperature which is not more than 15 F above the lowest recorded, and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 36.)

CRACK LENGTH USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. This method may be used to advantage as a check on the calculations made in the more exact manner.

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc.	½ to 3

temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone* is located at mid-height of a building, and that the temperature difference is 70 F, the following formulae may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \tag{1}$$

$$M_e = \sqrt{M^2 + 1.75\,b} \tag{2}$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height.
b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the high altitudes makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One

arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR INFILTRATION

Sensible Heat Loss

The heat required to warm cold outside air, which enters a room by infiltration, to the temperature of the room is given by the equation:

$$H_s = 0.24\,Q\,d\,(t_i - t_o) \tag{3}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t_i Btu per hour.
0.24 = specific heat of air.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_o, pounds per cubic foot.
t_i = room air temperature, degrees Fahrenheit.
t_o = outside air temperature, degrees Fahrenheit.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\,d\left(\frac{M_i - M_o}{7000}\right)L \tag{4}$$

where

H_l = heat required to increase moisture content of air leaking into building from M_o to M_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
M_i = vapor density of inside air, grains per pound of dry air.
M_o = vapor density of outside air, grains per pound of dry air.
L = latent heat of vapor at M_i, Btu per pound.

It is sufficiently accurate to use d = 0.075 lb, in which case equation 3 reduces to 5 and if the latent heat of vapor is assumed for general conditions as 1060 Btu per pound equation 4 reduces to 6.

$$H_s = 0.018\,Q\,(t_i - t_o) \tag{5}$$

$$H_l = 0.0114\,Q\,(M_i - M_o) \tag{6}$$

While a heating reserve must be provided to warm inleaking air on the windward side of a building, this does not necessarily mean that the

heating plant must be provided with a reserve capacity, since the inleaking air, warmed at once by adequate heating surface in exposed rooms, will move transversely and upwardly through the building, thus relieving other radiators of a part of their load. The actual loss of heat of a building caused by infiltration is not to be confused with the necessity for providing additional heating capacity for a given space. Infiltration is a disturbing factor in the heating of a building, and its maximum effect (maximum in the sense of an average of wind velocity peaks during the heating season above some reasonably chosen minimum)must be met by a properly distributed reserve of heating capacity, which reserve, however, is not in use at all places at the same time, nor in any one place at all times.

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PROBLEMS IN PRACTICE

1 ● What two natural forces cause infiltration?

a. Wind causes pressure to be exerted on one or two sides of a building with consequent movement of air through openings.

b. Temperature difference inside to outside causes a difference in density with consequent entrance of air at lower openings and exit of air at higher openings.

2 ● What is the neutral zone as applied to infiltration in buildings?

Due to temperature difference inside air tends to flow out of openings near the top of the building and is replaced by outside air coming in at lower openings. At some height no flow occurs in or out. This is referred to as the neutral zone and often is taken at the mid-height of the building but may be higher or lower than this depending on several factors.

3 ● In what type of structure is infiltration due to temperature difference of importance?

In tall open buildings, in multi-story buildings inadequately sealed between floors and in stair-wells for multi-story buildings the temperature difference effect is important.

4 ● What procedure is sometimes used to compensate for this temperature or chimney effect in stair-wells?

One rule is to place 50 percent of the calculated radiation in the bottom third, the normal amount in the middle third and the remainder in the top third.

5 ● Why is it customary to apply a correction factor to laboratory values before calculating infiltration in buildings?

Most, if not all, laboratory values have been determined by exerting a certain wind pressure across a single thickness of building construction. In actual building construction the pressure drop due to wind velocity takes place in two steps, one through the windward and the other through the leeward wall. Tests indicate that the leakage in actual construction will be about 80 per cent of laboratory values and therefore this factor has been applied in making up the tables in this chapter. The curves represent laboratory data as taken with no correction applied.

6 ● Is infiltration through walls of importance?

In the case of compound walls of good construction the heat loss due to infiltration is usually negligibly low. In the case of single thickness walls without building paper properly applied, the heat loss due to air leakage may be very high.

7 ● How are the wind velocities and outside temperatures selected for infiltration calculations in heating?

It is common practice to take the average wind velocity for the three coldest months and the temperature as 15 F above the coldest recorded by the Weather Bureau during the preceeding 10 years.

8 ● Show the probable combined effects of infiltration due to temperature difference and wind velocity on the ground floor and on the 15th floor of a 20-story building 200 ft high with a 12 mph wind blowing.

At the ground floor the effective velocity is increased to:

$$M_e = \sqrt{12^2 + 1.75 \times 100} = 17.8 \text{ mph}$$

At the 15th floor level it would be reduced to:

$$M_e = \sqrt{12^2 - 1.75 \times 50} = 7.5 \text{ mph}$$

9 ● What is the value of storm sash in reducing infiltration?

The reduction depends on the relative fit of the window and the storm sash. Fig. 4 indicates for storm sash buttoned on a reduction at 20 mph from about 52 cfh per foot of crack to 42 cfh. Fig. 5 indicates a reduction for a storm sash buttoned over a loosely fitted window from 185 to 90 cfh.

Chapter 7

HEATING LOAD

Heat Demand Design Factors, Method of Procedure, Inside and Outside Temperatures, Wind Velocity Effects, Auxiliary Heat Sources, Wall Condensation, Heat Loss Computation

TO design any system of heating, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be capable of maintaining the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

1. Outside temperature. 2. Rain or snow. 3. Sunshine or cloudiness. 4. Wind velocity.	*Outside Conditions (The Weather)*
5. Heat transmission of exposed parts of building. 6. Infiltration of air through cracks, crevices and open doors and windows. 7. Heat capacity of materials. 8. Rate of absorption of solar radiation by exposed materials.	*Building Construction*
9. Inside temperatures. 10. Stratification of air. 11. Type of heating system. 12. Ventilation requirements. 13. Period and nature of occupancy. 14. Temperature regulation.	*Inside Conditions*

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time, while desirable, is hard of attainment.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1.)

2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself. (See Table 2.)

3. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 5.)

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1 and 2.)

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 6.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions re-

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg Fahr	Type of Building	Deg Fahr
SCHOOLS		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. See Chapter 3.

quire, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 3. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H. V.E. Research Laboratory, is 66 deg.[1]

According to Fig. 6, Chapter 3, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat-loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of pitched roofs, unheated attics and top-floor ceilings Chapter 5.)

[1]See Chapter 3, p. 63.

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight-line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information[2].

Where ceiling heights are under 20 ft, it is common engineering practice to consider that the Fahrenheit temperature increases 2 per cent for each foot of height above the breathing line. This rule, sufficiently accurate for most cases, will give the probable air temperature at any given level for a room heated by direct radiation. Thus, the probable temperature in a room at a point 3 ft above the breathing line, if the breathing line temperature is 70 F, will be

$$(1.00 + 3 \times .02)\ 70 = 74.2 \text{ F.}$$

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters, air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling by a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing-line temperature may be used, provided the breathing-line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat-loss computations.

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[2]Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243).

Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

The outside temperature to be assumed in the design of any heating system is ordinarily not more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In the case of massive and well insulated buildings in localities where the minimum does not prevail for more than a few hours, or where the lowest recorded temperature is extremely unusual, more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau at the places listed.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND VELOCITY EFFECTS

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.
2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 5, by using a surface coefficient f_o for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 5 are based.

In a similar manner, the heat allowance for infiltration through cracks

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State	City	Average Temp., Oct. 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Mobile	58.9	−1	10.4	N
	Birmingham	53.8	−10	8.5	N
Ariz.	Phoenix	59.5	12	6.4	E
	Flagstaff	35.8	−25	7.8	SW
Ark.	Fort Smith	50.4	−15	8.1	E
	Little Rock	51.6	−12	8.7	NW
Calif.	San Francisco	54.2	27	7.6	N
	Los Angeles	58.5	28	6.3	NE
Colo.	Denver	38.9	−29	7.5	S
	Grand Junction	38.9	−21	5.3	NW
Conn.	New Haven	38.4	−15	9.7	N
D. C.	Washington	43.4	−15	7.1	NW
Fla.	Jacksonville	62.0	10	9.2	NE
Ga.	Atlanta	51.5	−8	12.1	NW
	Savannah	58.5	8	9.5	NW
Idaho	Lewiston	42.3	−23	5.3	E
	Pocatello	35.7	−28	9.6	SE
Ill.	Chicago	36.4	−23	12.5	W
	Springfield	39.8	−24	10.1	NW
Ind.	Indianapolis	40.3	−25	11.5	SW
	Evansville	45.1	−16	9.8	S
Iowa	Dubuque	33.9	−32	7.1	NW
	Sioux City	32.6	−35	11.6	NW
Kans.	Concordia	39.8	−25	8.1	S
	Dodge City	41.4	−26	9.8	NW
Ky.	Louisville	45.3	−20	9.9	SW
La.	New Orleans	61.6	7	8.8	N
	Shreveport	56.2	−5	8.9	SE
Me.	Eastport	31.5	−23	12.0	W
	Portland	33.8	−21	9.2	NW
Md.	Baltimore	43.8	−7	7.8	NW
Mass.	Boston	38.1	−18	11.2	W
Mich.	Alpena	29.6	−28	12.4	W
	Detroit	35.8	−24	12.7	SW
	Marquette	28.3	−27	11.1	NW
Minn.	Duluth	24.3	−41	12.6	SW
	Minneapolis	29.4	−33	11.3	NW
Miss.	Vicksburg	56.8	−1	8.3	SE
Mo.	St. Joseph	40.7	−24	9.3	NW
	St. Louis	43.6	−22	11.6	S
	Springfield	44.3	−29	10.8	SE
Mont.	Billings	34.0	−49	....	W
	Havre	27.6	−57	9.5	SW
Nebr.	Lincoln	37.0	−29	10.5	S
	North Platte	35.4	−35	8.5	W
Nev.	Tonopah	39.4	−10	10.0	SE
	Winnemucca	37.9	−28	8.7	NE
N. H.	Concord	33.3	−35	6.6	NW
N. J.	Atlantic City	41.6	−9	15.9	NW
N. Y.	Albany	35.2	−24	8.1	S
	Buffalo	34.8	−20	17.2	W
	New York	40.7	−14	17.1	NW

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Continued)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State or Province	City	Average Temp., Oct. 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
N. M.	Santa Fe	38.3	−13	7.8	NE
N. C.	Raleigh	50.0	−2	8.2	SW
	Wilmington	54.2	5	8.5	SW
N. Dak.	Bismarck	24.6	−45	9.1	NW
	Devils Lake	20.3	−44	10.6	W
Ohio	Cleveland	37.2	−17	13.0	SW
	Columbus	39.9	−20	12.0	SW
Okla.	Oklahoma City	47.9	−17	12.0	N
Oreg.	Baker	35.2	−24	6.9	SE
	Portland	46.1	−2	7.5	S
Pa.	Philadelphia	42.7	−6	11.0	NW
	Pittsburgh	41.0	−20	11.7	W
R. I.	Providence	37.2	−17	12.8	NW
S. C.	Charleston	57.4	7	10.6	SW
	Columbia	54.0	−2	8.1	NE
S. Dak.	Huron	28.2	−43	10.6	NW
	Rapid City	33.4	−34	8.2	W
Tenn.	Knoxville	47.9	−16	7.8	SW
	Memphis	51.1	−9	9.7	S
Texas	El Paso	53.5	−5	10.4	NW
	Fort Worth	55.2	−8	10.4	NW
	San Antonio	60.6	4	8.0	NE
Utah	Modena	36.3	−24	8.8	W
	Salt Lake City	40.0	−20	6.7	SE
Vt.	Burlington	31.5	−29	11.8	S
Va.	Norfolk	49.3	2	12.5	N
	Lynchburg	46.8	−7	7.1	NW
	Richmond	47.0	−3	7.9	SW
Wash.	Seattle	44.8	3	11.3	SE
	Spokane	37.7	−30	7.1	SW
W. Va.	Elkins	39.4	−28	6.6	W
	Parkersburg	42.6	−27	7.5	SW
Wis.	Green Bay	30.0	−36	10.4	SW
	La Crosse	31.7	−43	7.3	S
	Milwaukee	33.4	−25	11.5	W
Wyo.	Sheridan	30.7	−41	6.0	NW
	Lander	30.0	−40	5.0	SW
Alta.	Edmonton	23.0	−57	6.5	SW
B. C.	Victoria	43.9	− 1.5	12.5	N
	Vancouver	42.0	2	4.5	E
Man.	Winnipeg	17.5	−47	10.0	NW
N. B.	Fredericton	27.0	−35	9.6	NW
N. S.	Yarmouth	35.0	−12	14.2	NW
Ont.	London	32.6	−27	10.3	SW
	Ottawa	26.5	−34	8.4	NW
	Port Arthur	22.4	−37	7.8	NW
	Toronto	32.9	−26.5	13.0	SW
P. E. I.	Charlottetown	29.0	−27	9.4	SW
Que.	Montreal	27.8	−29	14.3	SW
	Quebec	24.2	−34	13.6	SW
Sask.	Prince Albert	15.8	−70	5.1	W
Yukon	Dawson	2.1	−68	3.7	S

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

and walls (Tables 1 and 2, Chapter 6) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 6).

In the past many designers have used empirical *exposure factors* which were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities. Recent investigations show, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

The exposure factor, which is still in use by many engineers, is usually taken as 15 per cent, and is added to the calculated heat loss on the side or sides exposed to what is considered the prevailing winter wind. There is a need for actual test data on this point, and pending the time when it can be secured, the question must be left to the judgment of the designing engineer. It should be remembered that the values of U in the tables in Chapter 5 are based on a wind velocity of 15 mph and that the infiltration figures are supposed to be selected from the tables in Chapter 6 to correspond to the wind velocities given in Table 2 of the present chapter.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method[3] for calculating the square feet of equivalent direct radiation required in a building. This method makes use of exposure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and the velocity of them.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is

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[3]See Standards of *Heating, Piping and Air Conditioning Contractors National Association.*

retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case Btu per hour = bhp $\times$ 2546, in which 2546 is the Btu equivalent of 1 hp-hour. In high-powered mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.415. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas gives off about 535 Btu per hour; and one cubic foot of natural gas gives off about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 3.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 3.)

WALL CONDENSATION

Condensation in the interior surfaces[4] of buildings may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power, and causes disintegration of roof structures not properly protected.

The prevalence of moisture on a surface is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature. It can be eliminated by (1) raising the surface temperature with increased air velocities passing over the surface, or adding a sufficient thickness of insulation, and (2) by lowering the humidity which is often not possible due to manufacturing processes.

The condensation of moisture within walls[5] is an important problem with may types of construction under adverse conditions. The tempera-

[4]Preventing Condensation on Interior Building Surfaces, by Paul D. Close (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 153).

[5]Condensation within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1938).

tures of the various parts of a wall are controlled by the type and amount of insulation used and the vapor densities in the corresponding sections are controlled by the type of vapor barriers installed. The transmission of heat and vapor through a wall should be considered together, and in most cases the proper combination of insulation and vapor barriers will eliminate the possibilities of condensation within walls. A consideration often overlooked in problems of condensation within walls is that a vapor barrier should be placed on the warm side and not on the cold side of a wall.

HEAT LOSS COMPUTATION EXAMPLE

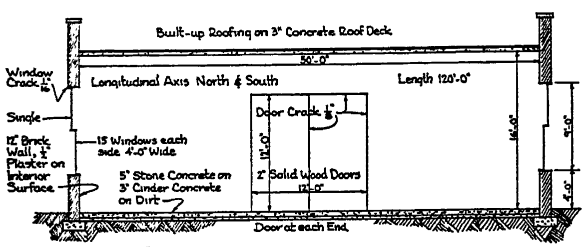


FIG. 1. ELEVATION OF FACTORY BUILDING

1. LOCATION..Philadelphia, Pa.

2. LOWEST OUTSIDE TEMPERATURE. (Table 2).. – 6 F

3. BASE TEMPERATURE: *In this example* a design temperature 10 F above lowest on record instead of 15 F is used. Hence the base temperature =

$$(-6 + 10) = +4 \text{ F}.$$

4. DIRECTION OF PREVAILING WIND (during Dec., Jan., Feb.)....................Northwest

5. BREATHING-LINE TEMPERATURE (5 ft from floor)..60 F

6. INSIDE AIR TEMPERATURE AT ROOF:

   The air temperature just below *roof* is higher than at the breathing line. Height of roof is 16 ft, or it is 16 – 5 = 11 ft above breathing line. Allowing 2 per cent per foot above 5 ft, or 2 × 11 = 22 per cent, makes the temperature of the air under the roof = 1.22 × 60 = 73.2 F.

7. INSIDE TEMPERATURE AT WALLS:

   The air temperature at the mean height of the *walls* is greater than at the breathing line. The mean height of the walls is 8 ft and allowing 2 per cent per foot above 5 ft, the average mean temperature of the walls is 1.06 × 60 = 63.6 F. By similar assumptions and calculations, the mean temperature of the *glass* will be found to be 64.2 F and that of the *doors* 61.2 F.

8. AVERAGE WIND VELOCITY (Table 2)..11.0 mph

9. OVER-ALL DIMENSIONS (See Fig. 1)..120 x 50 x 16 ft

10. CONSTRUCTION:

    Walls—12-in. brick, with ½-in. plaster applied directly to inside surface.

    Roof—3-in. stone concrete and built-up roofing.

Floor—5-in. stone concrete on 3-in. cinder concrete on dirt.

Doors—One 12 ft x 12 ft wood door (2 in. thick) at each end.

Windows—Fifteen, 9 ft x 4 ft single glass double-hung windows on each side.

11. TRANSMISSION COEFFICIENTS:

Walls—(Table 3, Chapter 5, Wall 2B) .... $U = 0.34$
Roof—(Table 11, Chapter 5, Roofs 2A and 3A) .... $U = 0.77$
Floor—(Table 10, Chapter 5, Floors 5A and 6A) .... $U = 0.63$
Doors—(Table 13B, Chapter 5) .... $U = 0.46$
Windows—(Table 13A, Chapter 5) .... $U = 1.13$

12. INFILTRATION COEFFICIENTS:

Windows—Average windows, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance. The leakage per foot of crack for an 11-mile wind velocity is 25.0 cfh. (Determined by interpolation of Table 2, Chapter 6.) The heat equivalent per hour per degree per foot of crack is taken from Chapter 6.

$25.0 \times 0.018 = 0.45$ Btu per deg Fahrenheit per foot of crack.

Doors—Assume infiltration loss through door crack twice that of windows or $2 \times 0.45 = 0.90$ Btu per deg Fahrenheit per foot of crack.

Walls—As shown by Table 1, Chapter 6, a plastered wall allows so little infiltration that in this problem it may be neglected.

13. CALCULATIONS: See calculation sheet, Table 3.

TABLE 3. CALCULATION SHEET SHOWING METHOD OF ESTIMATING HEAT LOSSES OF BUILDING SHOWN IN FIG. 1

PART OF BUILDING	WIDTH IN FEET	HEIGHT IN FEET	NET SURFACE AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIFF.	TOTAL BTU
North Wall:						
Brick, 1/2-in. plaster	50	16	656	0.34	59.6	13,293
Doors (2-in. wood)	12	12	144	0.46	57.2	3,789
1/8 in. Crack	1 pair doors		60	0.90	57.2	1,544[a]
West Wall:						
Brick, 1/2-in. plaster	120	16	1380	0.34	59.6	27,964
Glass (Single)	15 x 4	9	540	1.13	60.2	36,734
1/16 in. Crack	Double Hung Windows (15)		450	0.45	60.2	6,095[a]
South Wall	Same as North Wall					18,626
East Wall	Same as West Wall					70,793
Roof, 3-in. concrete and slag-surfaced built-up roofing	50	120	6000	0.77	69.2	319,704
Floor, 5-in. stone concrete on 3-in. cinder concrete	50	120	6000	0.63	5[b]	18,900
GRAND TOTAL of heat required for building in Btu per hour						517,442

[a]This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack will be used in computing infiltration for each side and each end of building.

[b]A 5 F temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground.

PROBLEMS IN PRACTICE

1 ● What is the relation between the sensible heat loss from a building and the heat required for humidification?

A house with a volume of 14,000 cu ft has a heat loss 120 Mbh for standard uninsulated frame construction and a 70 F temperature difference. Assuming a leakage rate of 1½ air changes per hour it would require about 10 Mbh to maintain a relative humidity of 45 per cent when the outside air is 0 F and 50 per cent relative humidity. By using an insulation such as rock wool, the sensible heat loss of this house may be reduced to approximately 77 Mbh. The insulation does not affect the humidification load, which now assumes greater importance.

2 ● What inside dry-bulb temperatures are usually assumed for: (a) homes, (b) schools, (c) public buildings?

Referring to Table 1:

a. 70 to 72 F.

b. Temperature varies from 55 to 75 F, depending on the room. Classrooms, for instance, are usually specified as 70 to 72 F.

c. 68 to 72 F.

3 ● How is the outside temperature selected for use in computing heat losses?

The outside temperature used in computing heat losses is generally taken from 10 to 15 F higher than the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In some cases where the lowest recorded temperature is extremely unusual, the design temperature is taken even higher than 15 F above the lowest recorded temperature.

4 ● What are the effects of wind movement on the heating load?

a. Wind movement increases the heat transmission of walls, glass, and roof; it affects poor walls to a much greater extent than good walls.

b. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves if such materials are at all porous.

5 ● Calculate the heat given off by eighteen 200-watt lamps.

$200 \times 18 \times 3.415 = 12{,}294$ Btu per hour.

6 ● A two-story, six-room, frame house, 28-ft by 30-ft foundation, has the following proportions:

Area of outside walls, 1992 sq ft.
Area of glass, 333 sq ft.
Area of outside doors, 54 sq ft.
Cracks around windows, 440 ft.
Cracks around doors, 54 ft.
Area of second floor ceiling, 783 sq ft.
Volume, first and second floors, 13,010 cu ft.
Ceilings, 9 ft high.

The minimum temperature for the heating season is −34 F, and the required inside temperature at the 30-in. level is 70 F. The average number of degree days for a heating season is 7851, and the average wind velocity is 10 mph, northwest.

The walls are constructed of 2-in. by 4-in. studs with wood sheathing, building paper, and wood siding on the outside, and wood lath and plaster on the inside. Windows are single glass, double-hung, wood, without weatherstrips. The second floor ceiling is metal lath and plaster, without an attic floor. The roof is of wood shingles on wood strips with rafters exposed. The area of the roof is 20 per cent greater than the area of the ceiling. Select values for the following: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for roof;

(e) U for ceiling and roof combined; (f) air leakage, cubic feet per hour per foot of window crack; (g) air leakage, cubic feet per hour per foot of door crack.

a. 0.25 (Table 5, Chapter 5).
b. 1.13 (Table 13, Chapter 5).
c. 0.69 (Table 8, Chapter 5).
d. 0.46 (Table 12, Chapter 5).
e. 0.31 (Equation 6, Chapter 5).
f. 21.4 (Table 2, Chapter 6).
g. 42.8, which is double the window leakage.

7 ● Using the data of Question 6, calculate the maximum Btu loss per hour for the various constructions, and show the percentage of the total heat which is lost through each construction described.

Assume 2 per cent rise in temperature for each foot in height. The average temperature will be 72.8 F for walls, doors, and windows, and 79.1 F for the second floor ceiling.

a. Outside walls	46,200 Btu loss	35.4 per cent of total
b. Glass	34,950 Btu loss	26.7 per cent of total
c. Doors	5,670 Btu loss	4.4 per cent of total
d. Second floor ceiling	24,050 Btu loss	18.4 per cent of total
e. Air leakage, windows	15,750 Btu loss	12.1 per cent of total
f. Air leakage, doors	3,865 Btu loss	3.0 per cent of total
Total	130,485 Btu loss	100.0 per cent of total

8 ● For the house in Question 6, place 1-in. insulation in the outside walls and second floor ceiling; k for insulation = 0.34. Use weatherstrip on doors and windows, and double glass on the windows; C = 0.55. Calculate or select the following values: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for combination of ceiling and roof; (e) air leakage, cubic feet per hour per foot of door crack; (f) air leakage, cubic feet per hour per foot of window crack.

a. 0.144
b. 0.55
c. 0.23
d. 0.16
e. 15.5
f. 31.0

9 ● Calculate the maximum Btu loss per hour and show the percentage loss by each channel for the house as insulated in Question 8.

a. Outside walls	26,650 Btu loss	35.1 per cent of total
b. Glass	17,000 Btu loss	22.4 per cent of total
c. Doors	5,670 Btu loss	7.4 per cent of total
d. Ceiling	12,420 Btu loss	16.4 per cent of total
e. Air leakage, windows	11,400 Btu loss	15.1 per cent of total
f. Air leakage, doors	2,795 Btu loss	3.6 per cent of total
Total	75,935 Btu loss	100.0 per cent of total

10 ● From the results of Questions 7 and 9, calculate the Btu saved and the percentage saved by each change in construction.

	Uninsulated	Insulated	Btu Saved	Per Cent Saved
a. Outside walls	46,200	26,650	19,550	42.3
b. Glass	34,950	17,000	17,950	51.4
c. Doors	5,670	5,670	0	0
d. Ceiling	24,050	12,420	11,630	48.3
e. Air leakage, windows	15,750	11,400	4,350	27.6
f. Air leakage, doors	3,865	2,795	1,070	27.7

11 ● From the results of Questions 7 and 9, calculate the heat loads per heating season in Btu and note the savings by better construction.

The 7851 degree days for the heating season multiplied by 24 hours, times the Btu loss per hour for 1 F drop in temperature gives the Btu load per heating season.

Saving = 262,000,000 − 152,500,000 = 109,500,000 Btu.

12 ● The dry-bulb temperature and the relative humidity at the ceiling of a mixing room in a bakery are 80 F and 60 per cent, respectively. The roof is a 4-in. concrete deck covered with built-up roofing. If the lowest outside temperature to be expected is −10 F, what thickness of rigid fiber insulation will be required to prevent condensation?

From Table 11, Chapter 5, U for the uninsulated roof = 0.72. From Table 2, Chapter 5, k for rigid fiber insulation = 0.33. From the psychrometric chart the dew-point of air at 80 F and 60 per cent relative humidity is 65 F. The ceiling temperature, therefore, must not drop below 65 F if condensation is to be prevented.

When equilibrium is established, the amount of heat flowing through any component part of a construction is the same for each square foot of area.

Therefore,

$$U\,[80 - (-10)\,] = 1.65\,(80 - 65)$$

where

U is the transmittance of the insulated roof.

Solving the equation, $U = 0.275$.

The resistance of the insulated roof $= \dfrac{1}{0.275} = 3.64.$

The resistance of the uninsulated roof $= \dfrac{1}{0.72} = 1.39.$

The resistance of the insulation = 3.64 − 1.39 = 2.25.

Resistance per inch of insulation $= \dfrac{1}{0.33} = 3.0.$

Since a resistance of 2.25 is required, and 1 in. of insulation has a resistance of 3, one inch will be sufficient to prevent condensation.

The same result might have been obtained by selecting an insulated 4-in. concrete slab having a U of less than 0.275 from Table 11, Chapter 5. This 4-in. concrete slab with 1-in. rigid insulation has a U of 0.23 which is safe.

Chapter 8

COOLING LOAD

Conditions of Comfort, Design Outside Temperatures, Components of Heat Gain, Normal Heat Transmission, Solar Heat Transmission, Sun Effect Through Windows, Heat Emission of Occupants, Heat Introduced by Outside Air, Heat Emission of Appliances

LOAD calculations for summer air conditioning are more complicated than heating load calculations for the reason that there are several more factors to be considered. Because of the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be exercised in determining their phase relationship in order that equipment of proper capacity may be selected to maintain specified indoor conditions.

CONDITIONS OF COMFORT

The conditions to be maintained in an enclosure are variable and depend upon several factors, especially the outside design conditions, duration of occupancy and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper effective temperature to be maintained is given in Chapter 3, where are also tabulated the most desirable indoor conditions to be maintained in summer for exposures over 40 min (see Table 2, Chapter 3).

DESIGN OUTSIDE TEMPERATURES

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 1. It will be noted that the temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system covering this range. The temperatures shown in Table 1 are in part based on available design conditions known to be successfully applied and for those localities where this information is lacking they are based on a study of the hourly temperatures in New York City from which factors were derived and applied to the average maximum dry- and wet-bulb temperatures for other cities. This study covered a twenty-year record of Weather Bureau temperatures. The design

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

State	City	Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Ala.	Birmingham	95	78	5.2	S
	Mobile	95	80	8.6	SW
Ariz.	Phoenix	105	76	6.0	W
Ark.	Little Rock	95	78	7.0	NE
Calif.	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo.	Denver	95	64	6.8	S
Conn.	New Haven	95	75	7.3	S
Dela.	Wilmington	95	78	9.7	SW
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	95	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	95	75	7.3	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	95	76	8.2	S
Ind.	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Kansas	Wichita	100	75	11.0	S
Ky.	Louisville	95	76	8.0	SW
La.	New Orleans	95	79	7.0	SW
Maine	Portland	90	73	7.3	S
Md.	Baltimore	95	78	6.9	SW
Mass.	Boston	92	75	9.2	SW
Mich.	Detroit	95	75	10.3	SW
Minn.	Minneapolis	95	75	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	100	76	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	95	67	7.3	SW
Nebr.	Lincoln	95	75	9.3	S
Nev.	Reno	95	65	7.4	W
N. H.	Manchester	90	73	5.6	NW
N. J.	Trenton	95	78	10.0	SW
N. Y.	Albany	92	75	7.1	S
	Buffalo	93	75	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	90	65	6.5	SE
N. C.	Asheville	90	75	5.6	SE
	Wilmington	95	79	7.8	SW
N. Dak.	Bismarck	95	73	8.8	NW
Ohio	Cleveland	95	75	9.9	S
	Cincinnati	95	78	6.6	SW
Okla.	Oklahoma City	101	76	10.1	S
Oreg.	Portland	90	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R. I.	Providence	93	75	10.0	NW
S. C.	Charleston	95	80	9.9	SW
	Greenville	95	76	6.8	NE
S. Dak.	Sioux Falls	95	75	7.6	S
Tenn.	Chattanooga	95	77	6.5	SW
	Memphis	95	78	7.5	SW

TABLE 1. DESIGN DRY AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Continued)

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Texas	Dallas	100	78	9.4	S
	Galveston	95	80	9.7	S
	San Antonio	100	78	7.4	SE
	Houston	95	78	7.7	S
	El Paso	100	69	6.9	E
Utah	Salt Lake City	95	67	8.2	SE
Vt.	Burlington	90	73	8.9	S
Va.	Norfolk	95	78	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	85	65	7.9	S
	Spokane	90	65	6.5	SW
W. Va.	Parkersburg	95	75	5.3	SE
Wisc.	Madison	95	75	8.1	SW
	Milwaukee	95	75	10.4	S
Wyo.	Cheyenne	95	65	9.2	S

temperatures given are not exceeded more than 5 to 8 per cent of the time during a cooling season of 1200 hours in June, July, August and September for an average year.

COMPONENTS OF HEAT GAIN

A cooling load determination is composed of five components which may be classified in the following manner:

1. Normal heat transfer through windows, walls, partitions, doors, floors, ceilings, etc.
2. Transfer of solar radiation through windows, walls, doors, skylights, or roof.
3. Heat emission of occupants within enclosures.
4. Heat introduced by infiltration of outside air or controlled ventilation.
5. Heat emission of mechanical, chemical, gas, steam, hot water and electrical appliances located within enclosures.

The components of heat gain, classified by source are further classified as sensible and latent heat gain.

The first two components fall into the classification of sensible heat gain, that is, they tend to raise the temperature of the air within the structure. The last three components not only produce sensible heat gain but they may also tend to increase the moisture content of the air within the structure.

Normal Heat Transmission

By normal heat transmission, as distinguished from solar heat transmission is meant the transmission of heat through windows, walls, partitions, etc. from without to interior of enclosure by virtue of difference between outside and inside air temperatures. This load is calculated in a

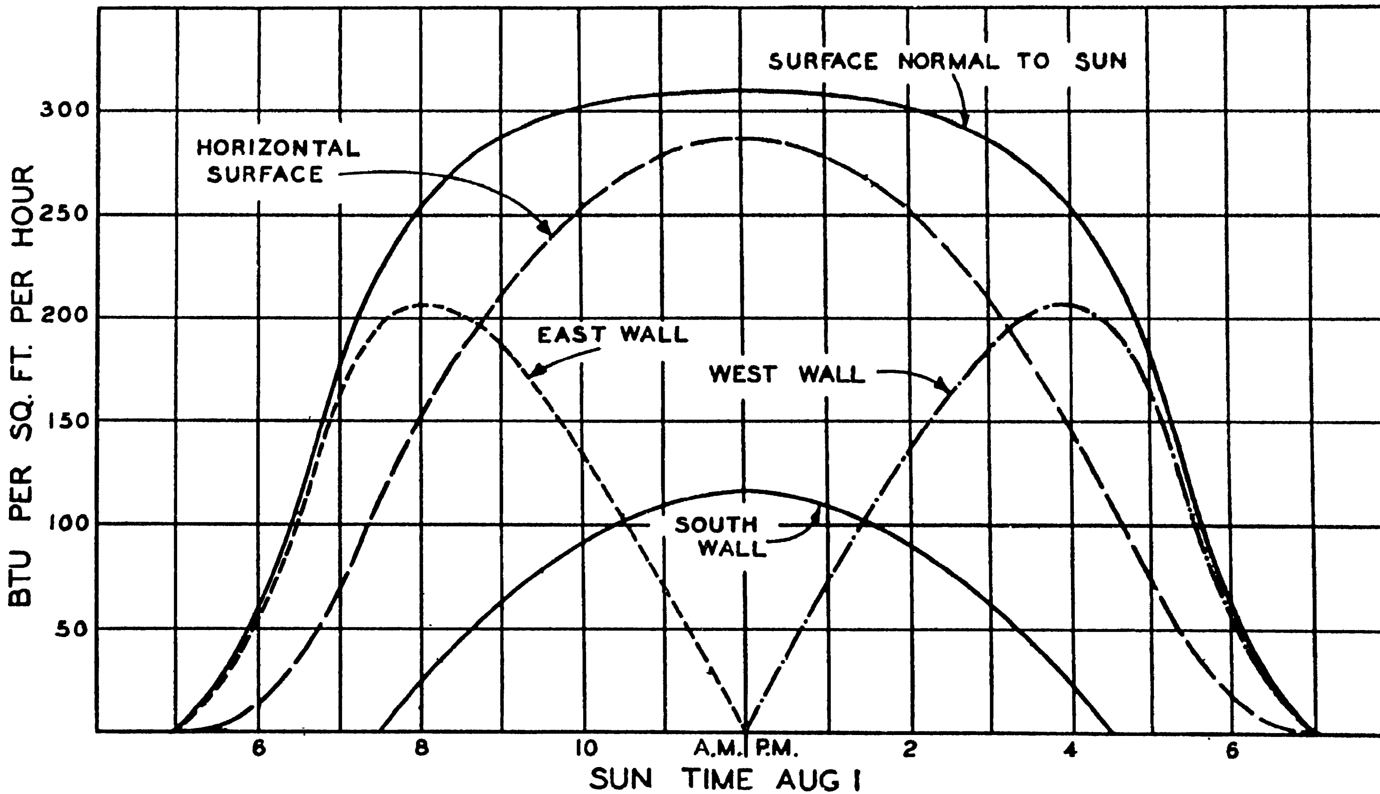


FIG. 1. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

manner similar to that described in Chapter 7 (except that flow of heat is reversed) by means of the formula:

$$H_t = A\,U\,(t_o - t) \qquad (1)$$

where

H_t = heat transmitted through the material of wall, glass, floor, etc., Btu per hour.
A = net inside area of wall, glass, floor, etc., square feet.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
U = coefficient of transmission of wall, glass, floor, etc., Btu per hour per square foot per degree Fahrenheit difference in temperature (Tables 3 to 13, Chapter 5).

Solar Heat Transmission

Calculations of the solar heat transmitted through walls and roofs are difficult to determine because of periodical character of heat flow and time lag due to heat capacity of construction.

The variation in solar intensity normal to sun in Btu per square foot per hour on a horizontal surface, and on east, west, and south walls is given in Fig. 1. The curves are drawn from A.S.H.V.E. Laboratory data obtained by pyrheliometer, are based on sun time and apply for a perfectly clear day on August 1 at a north latitude of 40 deg. A study of these curves discloses the periodic relationship and wide variation in solar intensity on various surfaces. It will be observed that both the roof and south wall radiation curves are in exact phase relationship with each other and that whereas the east and west wall radiation curves overlap those for roof and south wall, they do not overlap each other. This phase relationship has an important bearing on the cooling load. Failure to consider the periodical character of heat flow resulting from diurnal movement of the sun and the lag due to heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall, may result in a large error in load calculations.

The values of solar intensity appearing in Fig. 1 must not be confused with the actual heat transmission through the wall for much of this heat intensity on an outside surface is in part reflected and in part wiped off by convection to the outside air. A mathematical solution for the determination of solar heat transmission has been developed but the equations involved are too complex for practical application.[1] From results of this investigation and earlier studies,[2] the Research Laboratory has prepared Tables 2, 3, 4 and 5 which give the solar intensity (I) for various hours of the day against walls of various orientations and horizontal surfaces and the solar radiation (I_G) transmitted through windows of various orientations as well as skylights at various hours of the day. These values are shown for north latitudes from 30 to 45 deg.

It should be noted that the values for (I) represent the rate of solar intensity impinging against and not transmitted through walls and roofs whereas values for (I_G) represent actual rate of heat transmission through windows and skylights. Since the amount of solar intensity actually

[1]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. W. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 231).

[2]Absorption of Solar Radiation in its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 137).

TABLE 2. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE, AND THE RADIATION TRANSMITTED THROUGH GLASS FOR THE SAME ORIENTATIONS

For 30 Deg Latitude on the twenty-first of July, in Btu per sq ft per hour

Sun Time	Northeast		East		Southeast		South		Southwest		West		Northwest		Horizontal Surface	
	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G
4:59	0	0	0	0	0	0									0	0
5:00	1	0.9	1	0.9	0.3	0.2									0.01	0.005
6:00	47	41	51	44	24	17									9	5
7:00	136	106	160	140	90	68									68	47
8:00	151	122	205	177	136	105									147	116
9:00	127	91	189	156	140	104	8	1							214	182
10:00	79	47	141	103	122	85	31	12							265	231
11:00	21	6	78	45	85	52	45	21							296	261
12:00					36	15	50	24	36	15					305	269
1:00							45	21	85	52	78	45	21	6	296	261
2:00							31	12	122	85	141	103	79	47	265	231
3:00							8	1	140	104	189	156	127	91	214	182
4:00									136	105	205	177	151	122	147	116
5:00									90	68	160	140	136	106	68	47
6:00									24	17	51	44	47	41	9	5
7:00									0.3	0.2	1	0.9	1	0.9	0.01	0.005
7:01									0	0	0	0	0	0	0	0

TABLE 3. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS, AND A HORIZONTAL SURFACE, AND THE RADIATION TRANSMITTED THROUGH GLASS FOR THE SAME ORIENTATIONS

For 35 Deg Latitude on the twenty-first of July, in Btu per sq ft per hour.

Sun Time	Northeast		East		Southeast		South		Southwest		West		Northwest		Horizontal Surface	
	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G
4:46	0	0	0	0	0	0									0	0
5:00	9	8	9	8	3	2									0.01	0.007
6:00	67	58	72	63	35	26									15	8
7:00	142	120	174	152	103	78									77	53
8:00	150	120	209	181	145	114									151	120
9:00	118	83	191	157	154	118	26	8							214	181
10:00	60	32	143	104	139	101	55	27							264	230
11:00	2	0.1	75	43	103	67	72	41							291	256
12:00					55	28	78	46	55	28					300	265
1:00							72	41	103	67	75	43	2	0.1	291	256
2:00							55	27	139	101	143	104	60	32	264	230
3:00							26	8	154	118	191	157	118	83	214	181
4:00									145	114	209	181	150	120	151	120
5:00									103	78	174	152	142	120	77	53
6:00									35	26	72	63	67	58	15	8
7:00									3	2	9	8	9	8	0.01	0.007
7:14									0	0	0	0	0	0	0	0

TABLE 4. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE, AND THE RADIATION TRANSMITTED THROUGH GLASS FOR THE SAME ORIENTATIONS

For 40 Deg Latitude on the twenty-first of July, in Btu per sq ft per hour.

Sun Time	Northeast		East		Southeast		South		Southwest		West		Northwest		Horizontal Surface	
	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G
4:31	0	0	0	0	0	0									0	0
5:00	14	12	14	12	5	3									1	0.2
6:00	72	63	80	70	40	29									19	11
7:00	143	120	180	158	112	87									82	57
8:00	143	111	211	182	155	124	8	2							152	121
9:00	104	69	192	158	168	133	46	22							213	178
10:00	46	22	143	104	156	117	77	45							258	225
11:00			75	43	121	83	95	60	15	4					284	249
12:00					73	42	103	67	73	42					293	258
1:00					15	4	95	60	121	83	75	43			284	249
2:00							77	45	156	117	143	104	46	22	258	225
3:00							46	22	168	133	192	158	104	69	213	178
4:00							8	2	155	124	211	182	143	111	152	121
5:00									112	87	180	158	143	120	82	57
6:00									40	29	80	70	72	63	19	11
7:00									5	3	14	12	14	12	1	0.2
7:29									0	0	0	0	0	0	0	0

TABLE 5. SOLAR RADIATION IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE, AND THE RADIATION TRANSMITTED THROUGH GLASS FOR THE SAME ORIENTATIONS

For 45 Deg Latitude on the twenty-first of July, in Btu per sq ft per hour.

Sun Time	Northeast		East		Southeast		South		Southwest		West		Northwest		Horizontal Surface	
	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G	I	I_G
4:26	0	0	0	0	0	0									0	0
5:00	25	22	24	21	9	6									2	0 7
6:00	89	77	99	88	52	39									26	15
7:00	149	125	194	170	125	99									90	63
8:00	140	109	219	189	171	139	22	8							156	123
9:00	92	58	194	160	183	148	65	36							210	177
10:00	33	13	144	106	171	134	98	63							251	217
11:00			75	43	139	101	121	83	32	13					274	240
12:00					91	67	128	90	91	67					282	247
1:00					32	13	121	83	139	101	75	43			274	240
2:00							98	63	171	134	144	106	33	13	251	217
3:00							65	36	183	148	194	160	92	58	210	177
4:00							22	8	171	139	219	189	140	109	156	123
5:00									125	99	194	170	144	125	90	63
6:00									52	39	99	88	89	77	26	15
7:00									9	6	24	21	25	22	2	0.7

transmitted through a surface depends upon the nature of the exterior surface of wall or roof construction, it is necessary, in order to determine actual amount of solar heat transmission, to apply correction factors to the values of (I). Solar radiation factors and solar absorption coefficients have been determined[3] as indicated in Fig. 2 and Table 6 respectively.

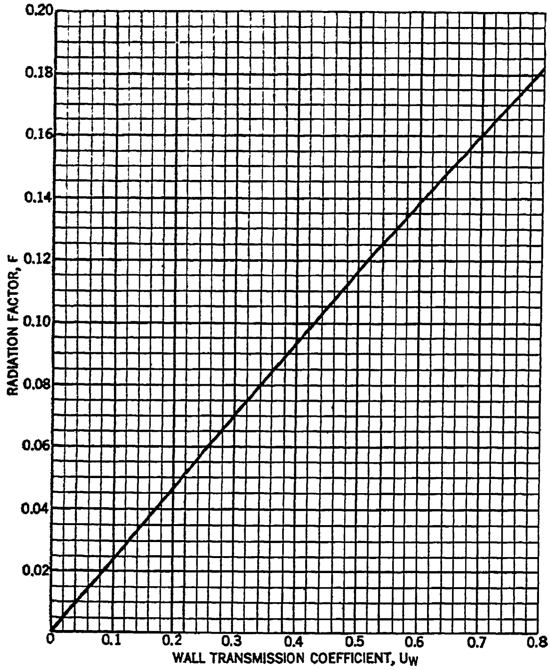


FIG. 2. SOLAR RADIATION FACTORS

The solar heat conduction through a wall or roof exposed to the sun may be expressed by the formula:

$$H_R = A\ F\ a\ I \tag{2}$$

where

H_R = Solar heat transmission, Btu per hour.

A = Area of wall or roof, square feet.

F = Percentage (expressed as a decimal) of the absorbed solar radiation which is transmitted to the inside (Fig. 2).

a = Percentage (expressed as a decimal) of the incident solar radiation which is absorbed by the surface (Table 6).

I = Actual intensity of solar radiation striking surface, Btu per hour per square foot (Tables 2, 3, 4 and 5).

[3]A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, L. Levine and F. O. Urban (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 327).

The total amount of heat conducted through a wall exposed to the sun is the sum of H_t and H_R from Formulas 1 and 2.

The calculation of heat transmission through walls and roofs does not take into consideration the heat capacity of the structure nor the consequent time lag in the transmission of heat. In the case of massive walls the time lag may amount to several hours[4]. Thus in many cases the wall transmission cannot be added directly to the cooling load from other sources because the peak of the wall transmission load may not coincide with the peak of the total cooling load and may even occur after the cooling system has been shut down for the day. The data in Table 7 were taken from A.S.H.V.E. research papers and whereas they result from a study of experimental slabs, they give an approximate idea of the time lag to be expected in various structures.

TABLE 6. SOLAR ABSORPTION COEFFICIENTS FOR DIFFERENT BUILDING MATERIALS

Surface Material	Absorption Coefficient (a)
Very Light Colored Surfaces White stone Very light colored cement White or light cream-colored paint	0.4
Medium Dark Surfaces Asbestos shingles Unpainted wood Brown stone Brick and red tile Dark-colored cement Stucco Red, green or gray paint	0.7
Very Dark Colored Surfaces Slate roofing Tar roofing materials Very dark paints	0.9

TABLE 7. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

Type and Thickness of Wall or Roof	Time Lag, Hours
2-in. pine	1½
6-in. concrete	3
4-in. gypsum	2½
3-in. concrete and 1-in. cork	2
2-in. iron and cork (equivalent to ¾-in. concrete and 2.15-in. cork)	2½
4-in. iron and cork (equivalent to 5½-in. concrete and 1.94-in. cork)	7¼
8-in. iron and cork (equivalent to 16-in. concrete and 1.53-in. cork)	19
22-in. brick and tile wall	10

[4]Loc. Cit. Notes 1 and 2.

Sun Effect Through Windows

Windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass through undiminished; only a small percentage (approximately 10 per cent) is reflected. This fact permits the solar heat gain through windows to be expressed by the simple formula:

$$H_G = A_G \, I_G \tag{3}$$

where

H_G = Solar radiation transmitted through a window, Btu per hour.

A_G = Area of glass, square feet.

I_G = Amount of solar radiation transmitted directly through the glass, Btu per hour per square foot (Tables 2, 3, 4 and 5).

Values for solar heat transmission through glass as determined by Formula 3 apply only to unshaded windows. Tests at the A.S.H.V.E. Research Laboratory[5] have determined the percentages of heat from solar radiation actually delivered to a room with bare windows and with various types of outdoor and indoor shading. The data in Table 8 are taken from these tests.

TABLE 8. SOLAR RADIATION TRANSMITTED THROUGH BARE AND SHADED WINDOWS

TYPE OF APPURTENANCE	FINISH FACING SUN	PER CENT DELIVERED TO ROOM
Bare window glass		97
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg	Aluminum	22

The percentage values in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar radiation plus glass transmission based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass is small or negative because the glass temperature is raised by the solar radiation absorbed. Therefore, in calculating the total heat gain through windows on the sunny sides of buildings, it is sufficiently accurate to determine the total cooling load due to the window, as the solar radiation times the proper factor from Table 8 and to neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air.

Although Table 8 shows that 97 per cent of the heat from solar radiation is delivered to a room through bare window glass, more recent tests[6]

[5]Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 93). Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 101).

[6]Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J. Wahl (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 53).

have indicated that in the case of a building having floors of high heat capacity such as concrete floors on which the solar radiation falls, approximately one-half of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load but is delivered back to the air in the building at a slow rate over a period of 24 hours or longer.

The maximum solar intensity on any surface is of limited duration as shown in Fig. 1. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be calculated as a steady load. Another point which should be noted is that the maximum solar radiation load on the east wall occurs early in the morning when the outside temperature is low.

In a paper[7] by the A.S.H.V.E. Research Laboratory it was shown that ordinary double strength window glass transmits no measurable amount of energy radiated from a source at 500 F or lower; that it transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively; and that it transmits 65.7 per cent of the radiation from an arc lamp, 76.3 per cent of the radiation from an incandescent tungsten lamp, and 89.9 per cent of the radiation from the sun. Thus, glass windows in a room constitute heat traps, which allow rather free transmission of radiant energy into the room from the sun to warm objects in it, but do not allow the transmission of re-radiated heat from these same objects.

Tests have been made which indicated that sunshine through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different times. In determining the total cooling load for a building if the time when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the times at which the sum of the loads on the different sides of the building is a maximum.

Heat Emission of Occupants

The heat and moisture given off by human beings under various states of activity are shown in Figs. 8 to 11 and Table 4 of Chapter 3. It will be observed that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large percentage of total load.

[7]Loc. Cit. Note 5.

Heat Introduced by Outside Air

An allowance must be made for the heat and moisture in the outside air introduced for ventilation purposes or entering the building through cracks, crevices, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapter 6. Information on the amount of outside air required for ventilation will be found in Chapter 3.

In the event the volume of air entering an enclosure due to infiltration exceeds that required for ventilation, the former should be used as a basis for determining the portion of the load contributed by outside air. Where volume of air required for ventilation exceeds that due to infiltration it is assumed that a slight positive pressure will exist within the enclosure with a resulting exfiltration instead of infiltration. In this case the air required for ventilation is used in determining outside air load.

The sensible heat gain resulting from the outside air introduced may be determined by the following formula:

$$H_s = 0.24 \times 60\, d_o\, Q\, (t_o - t) \qquad (4)$$

where

H_s = sensible heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering building, cubic feet per minute.
d_o = density of air, pounds of dry air per cubic foot at temperature t_o.
t_o = temperature of outside air, degrees Fahrenheit.
t = temperature of inside air, degrees Fahrenheit.

The total heat gain resulting from outside air introduced may be determined by the following formula:

$$H = 60\, d_o\, Q\, (h_o - h) \qquad (5)$$

where

H = total heat to be removed from outside air entering the enclosure, Btu per hour.
Q = volume of outside air entering enclosure, cubic feet per minute.
d_o = density of air, pounds of dry air per cubic foot of air (at temperature t_o).
h_o = heat content of mixture of outside dry air and water vapor, Btu per pound of dry air (at temperature t_o).
h = heat content of mixture of inside dry air and water vapor, Btu per pound of dry air (at temperature t).

The latent heat gain resulting from outside air introduced may be determined by the following formula:

$$H_l = H - H_s \qquad (6)$$

where

H_l = latent heat to be removed, Btu per hour.
H = total heat to be removed, Btu per hour.
H_s = sensible heat to be removed, Btu per hour.

Heat Emission of Appliances

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided

into three general classes of equipment or devices:

1. Electrical appliances.
2. Gas appliances.
3. Steam heating appliances.

In the first group may be found such devices as lights, motors, toasters, waffle irons, etc. The capacities of most electrical devices may be determined from the watt capacity indicated on their name plates. The Btu equivalent of heat generated per hour is determined by multiplying the watt capacity by 3.4 (one watthour is equivalent to 3.413 Btu).

The capacities of electric motors are usually expressed in terms of horsepower instead of watts. If the motor efficiency is known, the watts input may be calculated from the formula:

$$P = \frac{746\ (hp)}{n} \tag{7}$$

where

P = motor input, watts.
hp = motor load, horsepower.
n = motor efficiency (expressed as a decimal).

When the motor efficiency is not known the heat equivalent of electrical input can be approximately determined by applying data given in Table 9.

TABLE 9. HEAT GENERATED BY MOTORS

NAMEPLATE RATING HORSEPOWER	HEAT GAIN IN BTU PER HOUR PER HORSEPOWER	
	Connected Load in Same Room	Connected Load Outside of Room
1/8 to 1/2	4250	1700
1/2 to 3	3700	1150
3 to 20	2950	400

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 10.

Considerable judgment must be exercised in the use of data given in Table 10. Consideration must be given to time of day when appliances are used and the heat they contribute to the space at time of peak load. Only those appliances in use at the time of the peak load need be considered. Consideration must also be given to the way appliances are installed, whether products of combustion are vented to a flue, whether products of combustion escape into the space to be conditioned or whether appliances are hooded allowing part of the heat to escape through a stack connected with the hood. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that when the appliances are properly hooded with a positive fan exhaust system through the hood that 50 per cent of the heat will be conveyed up into the hood and the balance of 50 per cent will be dissipated in the space to be conditioned. Where latent as well as sensible heat is given off, it is usually safe to assume that all latent heat will be removed by a properly designed and operated vent or hood.

TABLE 10. HEAT GAIN FROM VARIOUS SOURCES

Source	Btu Per Hour		
	Sensible	Latent	Total
Electric Heating Equipment			
Electrical Equipment—Dry Heat—No Evaporated Water	100%	0%	100%
Electric Oven—Baking	80%	20%	100%
Electric Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Electric Lights and Appliances per Watt (Dry Heat)	3.4	0	3.4
Electric Lights and Appliances per Kilowatt (Dry Heat)	3413	0	3413
Electric Motors per Horsepower	2546	0	2546
Electric Toasters or Electric Griddles	90%	10%	100%
Coffee Urn—Large, 18 in. Diameter—Single Drum	2000	2000	4000
Coffee Urn—Small, 12 in. Diameter—Single Drum	1200	1200	2400
Coffee Urn—Approx. Connected Load per Gallon of Capacity	600	600	1200
Electric Range—Small Burner	*	*	3400
Electric Range—Large Burner	*	*	7500
Electric Range—Oven	8000	2000	10000
Electric Range—Warming Compartment	1025	0	1025
Steam Table—Per Square Foot of Top Surface	300	800	1100
Plate Warmer—Per Cubic Foot of Volume	850	0	850
Baker's Oven—Per Cubic Foot of Volume	3200	1300	4500
Frying Griddles—Per Square Foot of Top Surface	*	*	4600
Hot Plates—Per Square Foot of Top Surface	*	*	9000
Hair Dryer in Beauty Parlor—600 w	2050	0	2050
Permanent Wave Machine in Beauty Parlor—24-25 w Units	2050	0	2050
Gas Burning Equipment			
Gas Equipment—Dry Heat—No Water Evaporated	90%	10%	100%
Gas Heated Oven—Baking	67%	33%	100%
Gas Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Stove, Domestic Type—No Water Evaporated—Per Medium Size Burner	9000	1000	10000
Gas Heated Oven—Domestic Type	12000	6000	18000
Stove, Domestic Type—Heating Water—Per Medium Size Burner	5000	5000	10000
Residence Gas Range—Giant Burner (About 5½ in. Diameter)	*	*	12000
Residence Gas Range—Medium Burner (About 4 in. Diameter)	*	*	10000
Residence Gas Range—Double Oven (Total Size 18 in. x 18 in. x 22 in. High)	*	*	18000
Residence Gas Range—Pilot	*	*	250
Restaurant Range—4 Burners and Oven	*	*	100000
Cast-Iron Burner—Low Flame—Per Hole	*	*	100
Cast-Iron Burner—High Flame—Per Hole	*	*	250
Simmering Burner	*	*	2500
Coffee Urn—Large, 18 in. Diameter—Single Drum	5000	5000	10000
Coffee Urn—Small, 12 in. Diameter—Single Drum	3000	3000	6000
Coffee Urn—Per Gallon of Rated Capacity	500	500	1000
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table or Serving Table—Per Square Foot of Top Surface	400	900	1300
Dish Warmer—Per Square Foot of Shelf	540	60	600
Cigar Lighter—Continuous Flame Type	2250	250	2500
Curling Iron Heater	2250	250	2500
Bunsen Type Burner—Large—Natural Gas	*	*	5000
Bunsen Type Burner—Large—Artificial Gas	*	*	3000
Bunsen Type Burner—Small—Natural Gas	*	*	3000
Bunsen Type Burner—Small—Artificial Gas	*	*	1800
Welsbach Burner—Natural Gas	*	*	3000
Welsbach Burner—Artificial Gas	*	*	1800
Fish-tail Burner—Natural Gas	*	*	5000
Fish-tail Burner—Artificial Gas	*	*	3000
Lighting Fixture Outlet—Large, 3 Mantle 480 C.P.	4500	500	5000
Lighting Fixture Outlet—Small, 1 Mantle 160 C.P.	2250	250	2500
One Cubic Foot of Natural Gas Generates	900	100	1000
One Cubic Foot of Artificial Gas Generates	540	60	600
One Cubic Foot of Producer Gas Generates	135	15	150
Steam Heated Equipment			
Steam Heated Surface Not Polished—Per Square Foot of Surface	330	0	330
Steam Heated Surface Polished—Per Square Foot of Surface	130	0	130
Insulated Surface, Per Square Foot	80	0	80
Bare Pipes, Not Polished Per Square Foot of Surface	400	0	400
Bare Pipes, Polished Per Square Foot of Surface	220	0	220
Insulated Pipes, Per Square Foot	110	0	110
Coffee Urn—Large, 18 in. Diameter—Single Drum	2000	2000	4000
Coffee Urn—Small, 12 in. Diameter—Single Drum	1200	1200	2400
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table—Per Square Foot of Top Surface	300	800	1100
Miscellaneous			
Heat Liberated By Food per person, as in a Restaurant	30	30	60
Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops	100	200	300

*Per cent sensible and latent heat depends upon use of equipment; dry heat, baking or boiling.

GENERAL

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components. If the time when the maximum load occurs is not obvious the load should be calculated for various times of the day to determine the probable time at which the sum of the various component loads is a maximum.

Application of the foregoing data in determining cooling load requirements is illustrated in Example 1.

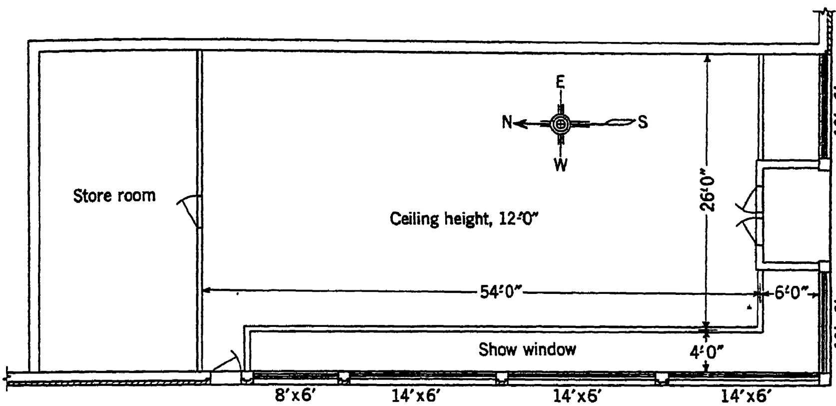


FIG. 3. PLAN DIAGRAM OF CLOTHING STORE

Example 1. Determine cooling load requirements for a clothing store illustrated in Fig. 3 and located in Cleveland, Ohio, Latitude 40 deg. This is a one-story building located on a corner and it faces south and west. Assume building on east and north sides conditioned.

Wall construction, 8 in. hollow tile, 4 in. brick veneer, plaster on walls, U = 0.33 (Table 4, Wall 38 B, Chapter 5).

Roof construction, 2 in. concrete, ½ in. rigid insulation, metal lath and plaster ceiling, U = 0.26 (Table 11, Wall 2 J, Chapter 5).

Floor, maple flooring on yellow pine, no ceiling below, U = 0.34 (Table 8, Wall 1 D, Chapter 5).

Partition, wood lath and plaster on both sides of studding, U = 0.34 (Table 6, Wall 77 B, Chapter 5).

Show windows, provided with awnings and thin panel partition at rear.

Front doors, 2 ft 6 in. x 7 ft (glass panelled), U = 1.13 (Table 13 A, Chapter 5).

Side door, 3 ft x 7 ft (solid, 1¾ in. thick), U = 0.51 (Table 13 B, Chapter 5).

Occupancy, 10 clerks, 40 patrons.

Lights, 4200 w.

Outside design conditions, dry-bulb 95 F; wet-bulb 75 F.

Inside design conditions, dry-bulb 80 F; wet-bulb 67 F.

Basement temperature, 85 F.

Store room temperature, 88 F.

Solution: The normal heat transmission through various surfaces shown in load calculations are determined by application of Formula 1.

It is quite obvious from the shape and exposure of this store that the maximum sun load will exist on the west wall. Since the west wall has a large glass area with a negligible time lag, the peak load may be expected at 4:00 p.m. at which time, from Table 4, I_G = 182. I_G for south glass at 4:00 p.m. is 2. Because of the small amount of solar radiation transmitted through the south glass, the transmission due to temperature difference has also been included. Assuming time lag in roof and walls to be 2 hours, the corresponding values for I for south and west walls and roof will be those shown in Table 4 for 2:00 p.m. They are respectively 77, 143 and 258. A time lag of 1 hour was assumed for the west door amounting to I = 192. By substituting these values in equations 2 and 3 the solar heat load is determined.

To determine the heat gain from the outside air it is necessary first to determine the volume of the outside air to be introduced. Since the show windows are sealed so as not to permit infiltration and since there are only three doors in this store through which infiltration can take place, it is obvious that infiltration of air will be a negligible quantity. The volume of the store is 21,600 cu ft. Good practice indicates that in a store of this character there should be a minimum of from 1 to 1½ outside air changes per hour. On a basis of 1½ air changes the volume of outside air to be introduced would be 32,400 cfh. By reference to Chapter 3 it will be noted that the minimum ventilation requirements are 10 cfm per person. On this basis the ventilation requirements would be 30,000 cfh. Since this will produce approximately 1½ outside air changes per hour, 30,000 cfh will be considered in this application.

To determine load imposed by occupants it will be found from Table 4, Chapter 3 that the average person standing at rest will dissipate 225 Btu sensible heat and 206 Btu latent heat per hour.

NORMAL TRANSMISSION LOAD:

SURFACE	DIMENSIONS	AREA SQ FT	U	TEMP. DIFF. DEG F	BTU PER HOUR
S Glass	2(2 ft 6 in. x 7 ft)+ 2(10 ft x 6 ft)	155	1.13	15	2,627
S Wall	(30 ft x 12 ft)−155	205	0.33	15	1,015
W Wall	(60 ft x 12 ft)−321	399	0.33	15	1,975
W Door	3 ft x 7 ft	21	0.51	15	161
Roof	60 ft x 30 ft	1800	0.26	15	7,020
Floor	26 ft x 54 ft	1404	0.34	5	2,387
N Partition	30 ft x 12 ft	360	0.34	8	979
Total					16,164

SUN LOAD:

SURFACE	DIMENSIONS	AREA SQ FT	F	a	I OR I_G	SHADE FACTOR	BTU PER HOUR
S Wall		205	0.078	0.7	77		862
S Glass		155			2	0.28	87
W Glass	3(14 ft x 6 ft)+ (8 ft x 6 ft)	300			182	0.28	15,288
W Door		21	0.118	0.7	192		333
W Wall		399	0.078	0.7	143		3,113
Roof		1800	0.062	0.9	258		25,914
Total							45,597

OUTSIDE AIR HEAT GAIN:

Sensible heat, $H_s = 0.24 \times 60\, d_o\, Q\, (t_o - t)$ (Formula 4).

$Q = 50 \times 10 = 500$ cfm.

Density of air at 95 F dry-bulb and 75 F wet-bulb for a barometric pressure of 29.92 in. is 0.07089 lb per cubic foot (Table 4, Chapter 1).

Dew-point of outdoor air is 66 F (psychrometric chart).

Partial pressure of vapor is 0.64378 in. Hg. (Pressure of saturated vapor at 66 F, Table 6, Chapter 1).

$W = 0.622 \left(\frac{e}{B - e}\right) = 0.622 \left(\frac{0.64378}{29.92 - 0.64378}\right) = 0.0137$ lb water vapor per pound dry air (Formula 5 a, Chapter 1).

$\frac{1}{1 \text{ plus } 0.0137} = 0.986$ lb dry air per pound outside air.

$d_o = 0.07089 \times 0.986 = 0.0699$ lb dry air per cubic foot outside air.

$H_s = 60 \times 500 \times 0.0699 \times 0.24\ (95\text{-}80) = 7549$ Btu per hour.

Total heat, $H = 60\ d_o\ Q\ (h_o - h)$ (Formula 5).

$h_o = 38.46$ Btu per pound dry air at 75 F wet-bulb (Table 6, Chapter 1).

$h = 31.51$ Btu per pound dry air at 67 F wet-bulb (Table 6, Chapter 1).

$H = 60 \times 0.0699 \times 500\ (38.46 - 31.51) = 14{,}574$ Btu per hour.

Latent heat gain from outside air = 14,574 − 7549 = 7025 Btu per hour.

PEOPLE HEAT GAIN:

$50 \times 225 = 11{,}250$ Btu per hour, sensible heat.

$50 \times 206 = 10{,}300$ Btu per hour, latent heat.

LIGHT HEAT GAIN:

$4200 \times 3.413 = 14{,}335$ Btu per hour.

SUMMARY:

Component of Load	Btu per Hour	
	Sensible	Latent
Normal Transmission Load	16,164	
Sun Load	45,597	
Outside Air Heat Gain	7,549	7,025
People Heat Gain	11,250	10,300
Light Heat Gain	14,335	
Total	94,895	17,325

TOTAL LOAD:

94,895 + 17,325 = 112,220 Btu per hour.

PROBLEMS IN PRACTICE

1 ● The outdoor and indoor temperatures are 90 F and 78 F, respectively. What is the amount of heat transmitted per hour through a 7 ft by 4 ft north window?

$H_t = 28 \times 1.13\ (90\text{-}78) = 380$ Btu per hour. (Equation 1, Chapter 8 and Table 13 A, Chapter 5).

2 ● a. If a restaurant has two 10 gal gas-heated coffee urns, what is the cooling load due to them?

b. What is the cooling load due to four 1350 w burners on an electric range?

a. $2 \times 10 \times 1000 = 20{,}000$ Btu per hour (Table 10).

b. $4 \times 1350 = 5400$ w = 5.4 kw.
$5.4 \times 3413 = 18{,}430$ Btu per hour (Table 10).

3 ● a. What is the maximum heat transmission for a flat roof located in Pittsburgh (latitude 40 deg) exposed to the sun with the outdoor and indoor tem-

perature 95 F and 80 F, respectively? The roof is of uninsulated 6 in. concrete with its underside exposed, and with a black upper surface.

b. What time of day will maximum cooling load due to the roof exist?

a. $H_t + H_R = [A\,U\,(t_o - t)] + [A\,FaI]$ (Formulas 1 and 2).

U for roof $= 0.64$ (Table 11, Wall 4 A, Chapter 5).

$F = 0.147$ (Fig. 2).

$a = 0.9$ (Table 6).

$I = 293$ (Table 4).

$H_t + H_R = [1 \times 0.64\,(95\text{-}80)] + [1 \times 0.147 \times 0.9 \times 293] = 48.5$ Btu per square foot per hour.

b. Maximum sun intensity occurs at noon (Table 4). Maximum effect in cooling load will occur at 3 p.m. (Table 7).

4 ● a. What is the maximum rate of heat delivered to a room through a bare window in the west wall of a building located in New Orleans (30 deg latitude)?

b. What time of day will it occur?

c. What will maximum rate be if window is protected by awning?

a. $H_G = A_G I_G$ (Formula 3).

$I_G = 177$ (Table 2).

$H_G = 1 \times 177 = 177$ Btu per square foot per hour.

b. At 4 p.m. (Table 2).

c. $177 \times 0.28 = 49.6$ Btu per square foot per hour (Table 8).

5 ● What is the heat gain per cubic foot of outside air introduced, under the following conditions if the barometric pressure is 29.5 in. Hg.?

Outdoor temperatures, 90 F dry-bulb and 75 F wet-bulb.

Inside temperatures, 78 F dry-bulb and 65 F wet-bulb.

Density of air at 90 F dry-bulb, 75 F wet-bulb and 29.5 in Hg. is 0.0705 lb per cubic foot (Table 4, Chapter 1).

Dew-point of outdoor air is 68.2 F (psychrometric chart).

Pressure of saturated vapor at 68.2 F is 0.6946 in. Hg. (Table 6, Chapter 1).

$W = 0.622 \left(\frac{e}{B - e}\right) = 0.622 \left(\frac{0.6946}{29.5 - 0.6946}\right) = 0.015$ lb water vapor per pound dry air (Formula 5a, Chapter 1).

$\frac{1}{1 \text{ plus } 0.015} = 0.985$ lb dry air per pound outside air.

$d_o = 0.0705 \times 0.985 = 0.06944$ lb dry air per cubic foot outside air.

Heat content outside dry air at 75 F wet-bulb = 38.46 Btu per pound.

Heat content inside dry air at 65 F wet-bulb = 29.96 Btu per pound.

Total heat, $H = 0.06944\,(38.46 - 29.96) = 0.59$ Btu per cubic foot.

6 ● A 7 × 4 ft west window is equipped with an inside aluminum finished Venetian blind which is adjusted to fully cover the window when the sun shines. The net glass area is 75 per cent of the total area of the window. What is the cooling load due to the window at 10 a.m. and 4 p.m.? Temperatures are: 10 a.m., outside 85 F and inside 77 F; 4 p.m., outside 95 F and inside 80 F. Latitude 40 deg.

10 a.m.: $H_t = 28 \times 1.13\,(85\text{-}77) = 253$ Btu per hour (Equation 1, and Table 13A, Chapter 5).

4 p.m.: $28 \times 0.75 = 21$ sq ft net glass area.

$H_G = 21 \times 182 \times 0.58 = 2217$ Btu per hour (Tables 4 and 8).

Chapter 9

FUELS AND COMBUSTION

Classification of Coal, Air for Combustion, Draft Required, Combustion of Anthracite, Firing Bituminous Coal, Burning Coke, Hand Firing, Classification and Use of Oil, Classification and Use of Gas

THE choice of fuel for heating is a question of economy, cleanliness, fuel availability, operation requirements, and control. The principal fuels to be considered are coal, oil, and gas.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it intc clear-cut types. Its chemical composition is some indication but coal: having the same chemical analysis may have distinctly different burnin; characteristics. Users are mainly interested in the available heat pe pound of coal, in the handling and storing properties, and in the burnin; characteristics. A description of the relationship between the qualitie of coals and these characteristics requires considerable space; a treatmen applicable to heating boilers is given in *U. S. Bureau of Mines Bulletin* 276

A classification of coals is given in Table 1, and a brief description of th kinds of fuels is given in the following paragraphs, but it should b recognized that there are no distinct lines of demarcation between th kinds, and that they graduate into each other:

Anthracite is a clean, dense, hard coal which creates very little dust in handling. is comparatively hard to ignite but it burns freely when well started. It is non-cakin; it burns uniformly and smokelessly with a short flame, and it requires little attention t the fuel beds between firings. It is capable of giving a high efficiency in the commc types of hand-fired furnaces. A tabulation of the quality of the various anthraci sizes will be found in *U. S. Bureau of Mines Report of Investigations* No. 3283.

Semi-anthracite has a higher volatile content than anthracite, it is not as hard ar ignites somewhat more easily; otherwise its properties are similar to those of anthracit

Semi-bituminous coal is soft and friable, and fines and dust are created by handling It ignites somewhat slowly and burns with a medium length of flame. Its caking pr perties increase as the volatile matter increases, but the coke formed is relatively wea Having only half the volatile matter content of the more abundant bituminous coals can be burned with less production of smoke, and it is sometimes called *smokeless co*

The term *bituminous coal* covers a large range of coals and includes many types havi distinctly different composition, properties, and burning characteristics. The coals ran from the high-grade bituminous coals of the East to the poorer coals of the West. Th caking properties range from coals which completely melt, to those from which t volatiles and tars are distilled without change of form, so that they are classed as no caking or free-burning. Most bituminous coals are strong and non-friable enough permit of the screened sizes being delivered free from fines. In general, they ign

easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

TABLE 1. CLASSIFICATION OF COALS BY RANK[f]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	Non-agglutinating[a]
II. Bituminous[c]	1 Low volatile bituminous coal	Dry F.C., 77 per cent or more and less than 86 per cent (Dry V.M., 23 per cent or less and more than 14 per cent)	
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 77 per cent (Dry V.M., 31 per cent or less and more than 23 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); and moist[b] Btu, 14,000[d] or more	
	4. High volatile *B* bituminous coal	Moist[b] Btu, 13,000 or more and less than 14,000[d]	
	5 High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[d]	Either agglutinating or non-weathering[e]
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[d]	Both weathering and non-agglutinating
	2. Sub-bituminous *B* coal	Moist Btu 9500 or more and less than 11,000[d]	
	3. Sub-bituminous *C* coal	Moist Btu 8300 or more and less than 9500[d]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a]If agglutinating, classify in low-volatile group of the bituminous class.

[b]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[c]Pending the report of the Subcommittee on Origin and Composition and Methods of Analysis, it is recognized that there may be non-caking varieties in each group of the bituminous class.

[d]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[e]There are three varieties of coal in the High-volatile *C* bituminous coal group, namely, Variety 1, agglutinating and non-weathering; Variety 2, agglutinating and weathering; Variety 3, non-agglutinating and non-weathering.

[f]Adapted from *A.S.T.M.* Standards on Coal and Coke, p. 68, *American Society for Testing Materials*, Philadelphia, 1934.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke.* The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

The sale of *petroleum cokes* for domestic furnaces has been small and is generally confined to the Middle West. They vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

In order to obtain perfect combustion a definite amount of air is required for each pound of fuel fired. A deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack.

Total Air Required. The theoretical amount of air required per pound of fuel for perfect combustion is dependent upon the analysis of the fuel;

TABLE 2. POUNDS OF AIR PER POUND OF FUEL AS FIRED

ANTHRACITE	COKE	SEMI-BITUMINOUS	BITUMINOUS	LIGNITE
9.6	11.2	11.2	10.3	6.2

however, for estimating purposes the theoretical air required for different grades of fuel may roughly be taken from Table 2. An excess of about 50 per cent over the theoretical amount is considered good practice under usual operating conditions.

The amount of excess air, based upon the laws of combustion, can be determined by its relation to the percentage of CO_2 (carbon dioxide) in the products of combustion. This relationship is shown by the curves (Fig. 1) for high and low volatile coals and for coke. In hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

Secondary Air. The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed, and diameter of firepot. The ratio of the secondary to the primary air increases with decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the firepot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not

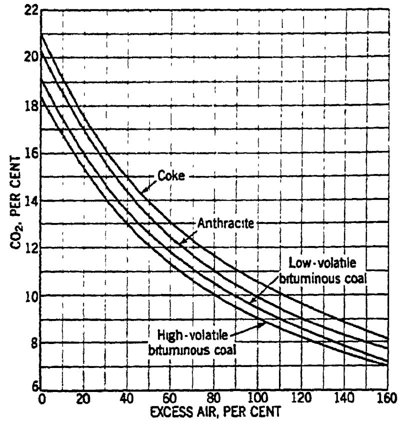
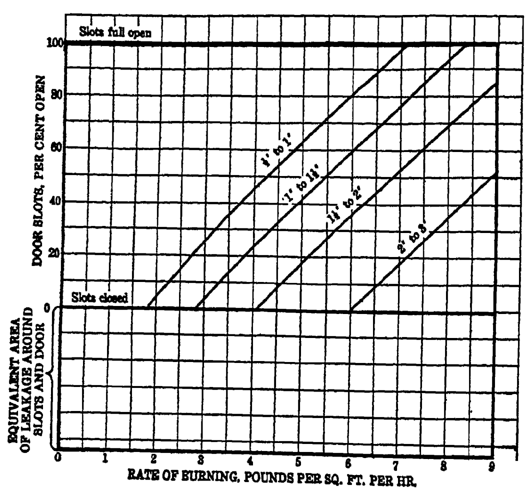


FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the



From U. S. Bureau of Mines.

FIG. 2. RELATIVE AMOUNT OF FIRE DOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

size of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

Fig. 2, taken from the *U. S. Bureau of Mines Report of Investigations* No. 2980, shows the relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning; these openings are with the ashpit damper wide open, and would be less if the available draft permits of its being partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.
2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.
3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.
4. For ordinary house operation, secondary air is needed after each firing for about one hour.

Draft Requirements

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning, and it is often possible to produce a higher rate by increasing the thickness of the fuel bed.

Combustion of Anthracite[1]

An anthracite fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and for best results this coal should be fired deeply.

[1]See reports published by *Anthracite Industries Laboratory*, Primos, Pennsylvania.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for firepots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater. A very satisfactory method of firing pea coal consists of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open, the cold-air check closed, and by controlling the fire with the air-inlet damper only. Pea size can also be fired in layers with stove or egg size anthracite and its use in this manner will reduce the fuel costs and attention required.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent an explosion of gas within the firepot, which in some cases depending upon the thickness of the bed of fresh coal is severe enough to blow open the doors and dampers of the furnace. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For best results and a higher degree of convenience, domestic stokers are used.

No. 2 buckwheat anthracite, or rice size, is used only in domestic stokers. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Bituminous Coal

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire door or by opening the fire door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the firebox, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the firebox and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the firebox so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The fuel bed should be carried as deep as the size of fuel and the available draft permit, in order to have as much coked fuel as possible for pushing to the rear of the firebox at the time of firing. A deep fuel bed allows the longest firing intervals.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since soft coal burns more rapidly than hard coal and with less draft. Soft coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

Semi-bituminous coal is fired as bituminous coal, and because of its caking characteristics it requires practically the same attention. The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

Burning Coke

Coke is a very desirable fuel and usually will give satisfaction as soon as the user learns how to control the fire. Coke ignites and burns very rapidly with less draft than anthracite coal. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds more rapidly than an anthracite fire to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. A deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. Since coke weighs only about half as much as anthracite per cubic foot only about half as much can be put in the firepot, so it will be necessary to fire oftener. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large firepots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

Dustless Coal

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with a solution of calcium chloride or a mixture of calcium and magnesium chlorides. Both these salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery and in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is sometimes treated at the mine, but more usually by the local distributor just before delivery. The solution is sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size.

Pulverized Coal

Installations of pulverized coal burning plants in heating boilers are of the unit type, in which the pulverized coal is delivered into the furnace immediately after grinding, together with the proper amount of preheated

air. With this apparatus, where the necessary furnace volume is obtainable, high efficiencies can be obtained.

A 150-hp boiler has generally been considered the smallest size for which pulverized fuel is feasible. Complications are introduced if an installation with a single boiler has to take care of very light loads.

Hand Firing

Hand firing is the oldest and the most widely used method of burning coal for heating purposes. To keep the fuel bed in proper condition where hand firing is used, the following general rules should be observed:

1. Remove ash from fuel bed by shaking the grates whenever fresh fuel is fired. This removes ash from the fire, enables the air to reach the fuel, and does away with the formation of clinker which is melted ash.

2. Supply the boiler with a deep bed of fuel. Nothing is gained by attempting to fire a small amount of fuel. A deep bed of fuel secures the most economical results.

3. Remove ash from ashpit at least once daily. Never allow ash to accumulate up to the grates. If the ash prevents the air from passing through, the grate bars will burn out and much clinker trouble will be experienced.

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires practically no combustion space.

CLASSIFICATION OF OILS

Uniform oil specifications were prepared in 1929 by the *American Oil Burner Association*, in cooperation with the *American Petroleum Institute*, the *U. S. Bureau of Standards*, the *American Society for Testing Materials* and other interested organizations. Oil fuels were classified into six groups, as indicated by Table 3. When these specifications were prepared, it was generally accepted that the first three grades were adapted to domestic use, while the last three were suitable only for commercial and industrial burners.

Today domestic installations may use No. 4 oil of the so-called heavy oil group, when and if said oil very closely follows the specifications of No. 3. Up-to-date listing by the *Underwriter's Laboratories* should be referred to before a No. 4 grade of fuel is used which merely meets *Commercial Standards CS 12-35*.

Since the specifications as originally drawn provide for maximum limits only for the several grades, this differentiation has not proved stable. Realizing how unsatisfactory it is to have specifications which permit the

TABLE 3. COMMERCIAL STANDARD FUEL OIL SPECIFICATIONS[a]

A. Detailed Requirements for Domestic Fuel Oils[b]

Grade of Oil	Approx. Btu per Gal.[c]	Flash Point		Water and Sediment, Maximum	Carbon Residue Maximum	Distillation Test		Viscosity Maximum
		Min.	Max.			Max.	Min.	
No. 1 Domestic Fuel Oil A light distillate oil for use in burners requiring a high grade fuel.	139,000	100 F or legal	150 F	0.05%	0.02%	10% point 420 F End point 600 F		
No. 2 Domestic Fuel Oil A medium distillate oil for use in burners requiring a high grade fuel.	141,000	125 F or legal	190 F	0.05%	0.05%	10% point 440 F 90% point 620 F	End point 600 F	
No. 3 Domestic Fuel Oil A distillate fuel oil for use in burners where a low viscosity oil is required.	143,400	150 F or legal	200 F	0.1%	0.15%		90% point 620 F	Saybolt Universal at 100 F 70 seconds

[a]Adapted from "Fuel Oils," p. 2, *U. S. Department of Commerce, Bureau of Standards, Commercial Standard CS12-35*, Washington, 1935.

[b]Pour Point Maximum is 15 F. Lower or higher pour points may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 0 F under any conditions.

[c]Government specifications do not give Btu per gallon, but they are noted here for information only.

B. Detailed Requirements for Industrial Fuel Oils[d]

Grade of Oil	Approx. Btu per Gal.[c]	Flash Point,		Water and Sediment, Maximum	Ash Maximum	Viscosity, Maximum
		Min.	Max.			
No. 4. Industrial Fuel Oil An oil known to the trade as a light fuel oil for use in burners where a low viscosity industrial fuel oil is required.	144,500	150 F	See Note[e]	1.0%	0.1%	Saybolt Universal at 100 F 500 seconds
No. 5 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "B" for burners adapted to the use of industrial fuel oil of medium viscosity.	146,000	150 F		1.0%	0.15%	Saybolt Furol at 122 F 100 seconds
No. 6 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "C" for burners adapted to oil of high viscosity	150,000	150 F		2.0%		Saybolt Furol at 122 F 300 seconds

[d]Pour point may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 15 F under any conditions.

[e]Whenever required, as for example in burners with automatic ignition, a maximum flash point may be specified. However, these specifications shall not require a flash point less than 250 F under any conditions.

substitution of one grade for another, the *U. S. Bureau of Standards* in cooperation with the *American Society for Testing Materials* is figuring on a new set of specifications providing for definite limits for each grade. When these specifications are adopted, it is expected that the *National Board of Fire Underwriters* will retest all burners using oils of the maximum specifications for the grade so that if a burner is approved for a certain grade it will burn any oil meeting the specifications for that particular grade.

Several burners adapted to industrial use have recently been listed for automatic operation with No. 5 oil. Usually oils No. 5 or 6 require preheating for proper operation, but where conditions are favorable, No. 5 can be used without the equipment that this entails.

There are two reasons for the trend to lower grades of oil. While the lighter oils contain slightly more heat units per pound, the weight per gallon increases more rapidly than the decrease in heat units per pound, and oil is bought by the gallon. As a consequence, while a No. 1 oil may contain 139,000 Btu per gallon, oil No. 5 may test 146,000 Btu per gallon, or 6 per cent more. Usually there is a differential of 3 to 4 cents between the No. 1 and No. 5 oils, so that the economy of buying the heavier fuels is apparent; there remains the economic utilization of the heat content of the heavier oils.

The cost of oil fuel is dependent also upon the amount that can be delivered at one time, and the method of delivery. Common practice has split the tank of the truck delivering oils for domestic use into compartments of 150 to 500-gal capacity, and these *unit dumps* are made the basis of price. Where a truck can be connected to a storage-tank *fill* and quickly discharge its oil by pump, the price obviously can be less than where a smaller quantity must be drawn off in 5-gal cans and poured. For similar reasons an installation that can be supplied from a tank car on a siding provides for a lower unit fuel cost than one where the oil must be trucked, even in the large trucks holding 2,000 gal or more that are used for distributing the heavier oils.

GAS CLASSIFICATION

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes, and is often designated as *city gas.*

When gas is burned a large amount of water vapor is produced as one of the products of combustion. This ordinarily escapes up the chimney, carrying away with it a certain amount of heat. However, when the heat value of gas is determined in an ordinary calorimeter, this water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the gas. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heat value of gas is specified. The heat that is reclaimed by the condensation of the water vapor amounts to about 10 per cent of the total heat value. It is impractical to utilize the entire higher heat value of the gas in any house-heating

appliance, because to do so it would be necessary to cool the products of combustion down below their dew point, which is ordinarily in the neighborhood of 130 F.

The actual dew point in the chimney is different from the theoretical value because excess air is admitted not only at the burner but also at the backdraft diverter which lowers the dew point.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 700 to 1500 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 4 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 4 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different dis-

TABLE 4. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS, BASED ON GAS AT 60 F AND 30 IN. HG.

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (Deg Fahr)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1087	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	967	873	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1232	1120	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carburetted water gas	536	496	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	134	124	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

tricts, nor at successive times in the same district. There are limits to the variation allowable, because the specific gravity of the gas depends on its composition, and too great a change in the specific gravity necessitates a change in the adjustment of the burners of small appliances.

Table 4 shows that a large proportion of the products of combustion when gas is burned may consist of water vapor, and that the greater the proportion of water vapor, the lower the maximum attainable CO_2 by gas analysis. The table also shows that a low calorific value does not necessarily mean a low flame temperature since, for example, natural gas has a theoretical flame temperature of 3600 F and blue water gas of 3800 F, although it has a calorific value less than one third that of natural gas.

The quantity of air given in Table 4 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

PROBLEMS IN PRACTICE

1 ● Differentiate between the general characteristics of hard and soft coals.

Hard coals contain fixed carbon in large proportions and in addition more ash is present especially in the smaller sizes. Soft coals have an increasing percentage of carbon in combination with hydrogen which is volatile and will distill off under high temperature, producing smoke.

2 ● Name several important properties of coal from a utilization standpoint.

a. Caking tendency, whether none, weak, or strong.
b. Quantity of volatile matter.
c. Friability.
d. Fusibility of the ash.

3 ● What are the main data commonly available that fix the qualities of coal, and do these tell the whole story?

a. Calorific value, Btu per pound.
b. Proximate analysis giving percentages of moisture, volatile matter, fixed carbon, ash, and sulphur.
c. Temperature at which the ash softens.
d. Screen sizes.

Other important qualities not usually given are the friability of the coal, its caking tendency, and the qualities of the volatile matter. The percentage of ash and its fusion temperature do not tell how the ash is distributed or how much of it is less fusible lumps of slate or shale.

4 ● Are there available complete and sufficient data on gas and oils to fix their burning properties and furnace requirements?

Yes. Because gas and oils are of simple and uniform composition, data are available to fix their burning properties and furnace requirements, but the ability to control their combustion is somewhat less determinable.

5 ● What effect does moisture in fuels have on their efficiency?

With any solid fuel, latent and sensible heat are lost at the stack when moisture is dried out of the fuel in burning, and when its hydrogen is burned. Therefore, such fuels as sub-bituminous coal and lignite, which are high in moisture content, have a low efficiency. However, these efficiencies may be improved if the stack gases are cooled to room temperature, by heating the feed water, for example.

6 ● What are the advantages of a sized fuel for heating furnaces?

Because a sized fuel encourages a more uniform flow of air through the bed, the burning will be more uniform, and the bed will be less liable to develop holes and will require less attention. Uniformity of fuel size is more desirable as the area of the bed becomes smaller; it is less important with fuels that cake, but with sized fuels the caking will be more uniform and the air flow through the bed will be steadier. In addition, ash and pieces of slate are less likely to be segregated and to form lumps of clinker.

7 ● Does the size of a fuel affect the quantity of air required to burn it at a given rate?

The total air required to give the same gas analysis at the stack is independent of the size of the fuel burned, but for non-caking fuels the ratio of the air passing through the fuel bed to the total air entering the burner base decreases, for the same thickness of bed, as the size of the fuel becomes smaller; this decrease is very rapid for sizes less than one inch. For coals that cake, this ratio will depend on the way the caked bed is broken up and on the size of the resulting pieces.

8 ● Is the volatile matter which is given off when coals are burned of the same nature in all coals?

No. The products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash-free and moisture-free coal, increasing amounts of oils and tars are given up. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it is low in the sub-bituminous coals and in lignite.

9 ● Is smoke a primary product in the burning of fuels?

Visible smoke may include very small particles of carbon, oil, tar, water (condensed steam), and ash. Of these, the oils, tars, and ash are mainly primary products, and the water is partly primary. The carbon, which usually comprises the greater part of the smoke, results from the breaking up by heat of oils, tars, and such gases as methane, so it may be considered a secondary product.

10 ● Is the sulphur in coals detrimental to combustion?

Not so far as is known, but its complete combustion gives only 25 per cent as much heat as is given by the same weight of carbon. Sulphur is undesirable because it causes corrosion of flues and stacks, and also because its gases pollute the atmosphere, and damage buildings and vegetation.

11 ● How do deposits of soot on the surfaces of a boiler or heater affect the quantity of fuel burned?

There are two effects. The soot acts as an insulating layer over the surface and reduces the heat transmission to the water or air; the *Bureau of Mines Report of Investigations* No. 3272 shows that the loss of seasonal efficiency is not as great as has been believed and should not be over 6 per cent because the greater part of the heat is transmitted through the firepot. The soot clogs the passages and reduces the draft; the loss of efficiency from this action may be much more, and also the lack of draft results in unsatisfactory heating.

Chapter 10

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Chimney Construction, Chimneys for Gas Heating

THE design and construction of a chimney is so important a part of the heating engineer's work that a general knowledge of draft characterictics and calculations is essential.

Draft, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, *viz*, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick, or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system, although inherently a heat balance loss.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above

their normal rating. Natural draft systems have been, and are still being, employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above the normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical

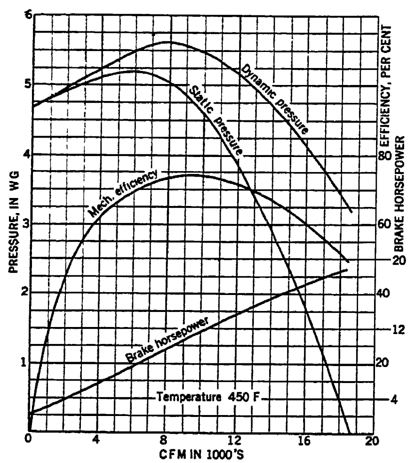


FIG. 1. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. Mechanical draft includes the induced and Venturi types of draft systems in which the pressure difference is the result of a suction, and also the forced draft system in which the pressure difference is the result of a blowing. Mechanical draft systems tend to produce a vacuum or a plenum, as the system used in its production creates a pressure difference below, or above, atmospheric

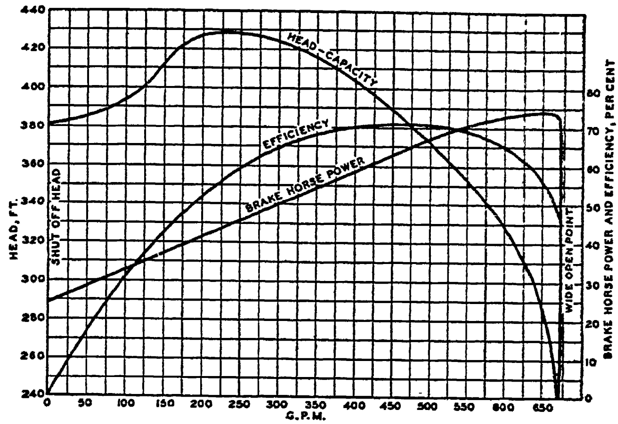


FIG. 2. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

pressure, respectively. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

Draft Control

To obtain the maximum efficiency of combustion, a definite minimum supply of air to the combustion chamber must be maintained. To provide this condition, it is necessary to have some mechanical means of draft control or adjustment, because of variable wind velocities, fluctuations in atmospheric temperatures and barometric pressures, and their effect upon draft.

For this purpose there are various mechanical devices which automatically control the volume of air admitted to the combustion chamber. Mechanical draft regulators designed to control or adjust draft, should

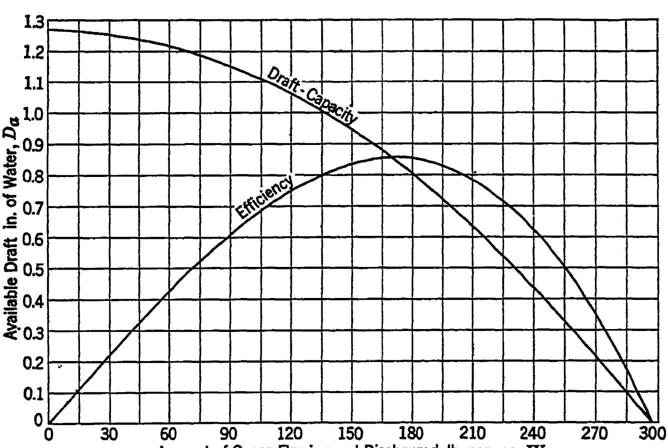


FIG. 3. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

not be confused with mechanical draft systems that *create* draft mechanically, but which must also be automatically controlled.

The use of such a device to provide a more uniform and dependable control of draft than could be maintained by manually operated dampers, will produce better combustion of fuel. This higher efficiency of combustion together with the reduced heat losses up the chimney by reason of decreased gas velocity, results in fuel economy, with consequent lower costs of plant operation.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it may be advantageous to compare its general operating characteristics with those of a centrifugal pump and also of a centrifugally-induced draft fan, there being a similarity among the three. Figs. 1, 2 and 3 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to the head-capacity curve of the pump and also to the dynamic-head capacity curve of the fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft.* When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the total *available draft.* The general equation for this net total available draft intensity of a natural draft chimney with a circular section is as follows:

$$D_a = 2.96HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{D^5B_oW_c} \qquad (1)$$

where

D_a = available draft, inches of water.
H = height of chimney above grate bars, feet.
B_o = barometric pressure corresponding to altitude, inches of mercury.
W_o = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.
W_c = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
T_o = absolute temperature of atmosphere, degrees Fahrenheit.
T_c = absolute temperature of chimney gases, degrees Fahrenheit.
W = amount of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.
f = coefficient of friction.
L = length of friction duct of the chimney, feet.
D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_o = 29.92 in. of mercury; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 1 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^6 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 3 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal

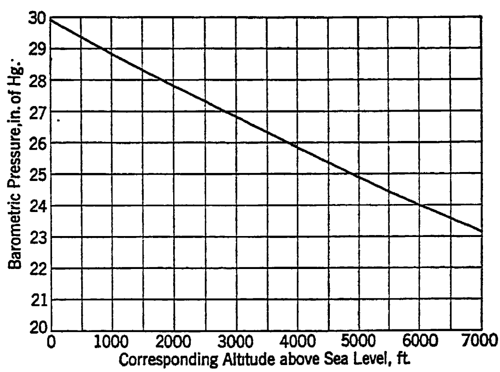


FIG. 4. RELATION BETWEEN BAROMETRIC PRESSURE AND ALTITUDE

pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 3.

In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure* varies inversely as the altitude of the plant above sea level Fig. 4 gives the barometric pressure corresponding to various elevations as computed from the equation:

$$E_1 = 62{,}737 \log_{10} \frac{29.92}{B_o} \qquad (2$$

where

E_1 = altitude of plant above sea level, feet.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_C = 0.131 CO_2 + 0.095\, O_2 + 0.083\, N_2 \tag{3}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_C may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are here disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* does not vary appreciably from the gas temperature as it leaves the breeching and enters the chimney. For average operating conditions, the chimney gas temperature will vary between 500 F and 650 F except in the case when economizers and recuperators are used, when the temperature will vary between 300 F and 450 F. If a chimney has been properly constructed, properly lined and has no air infiltration due to open joints, the temperature of the gases throughout the chimney will not differ appreciably from the foregoing figures. In most up-to-date heating plants, the temperature may be read from instruments or ascertained from a pyrometer. The analysis of this section is predicated on the assumption of constant gas temperature and no air infiltration throughout the height of the chimney.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

However, much to be recommended as an alternate method is the practise of separately determining duct friction factors as a function of the flow conditions, specifically as a function of the Reynolds number and the relative duct roughness. The Reynolds criterion is based on the physical properties of the gas, the duct dimensions, and the gas velocity. The gas velocity for a chimney is usually well above the critical velocity. It is likely that this procedure of using a separately determined variable friction factor for chimney flow will give results that are to be preferred over those based on a set constant.

The Reynolds number, a dimensionless ratio, may be stated as follows:

$$C_r = \frac{DV\rho}{\mu} \tag{4}$$

where

D = chimney diameter, feet.
V = velocity of hot gas, feet per second.
ρ = mass density of the chimney gas per cubic foot.
μ = viscosity of the gas in pounds-second per square foot taken at the gas temperature.

In another form:

$$C_r = \frac{1.27\, W}{D\mu g} = \frac{0.0396\, W}{D\mu} \tag{5}$$

where

W = weight of gas passed per second.
g = acceleration of gravity.

The value of μ for chimney gases is usually taken as that of air or nitrogen, and for the variation of μ with temperature, the Sutherland equation may be employed as follows, giving μ in pounds-second per square foot:

$$\mu = \mu_o \left[\frac{273 + C}{T_c + C}\right] \left[\frac{T_c}{273}\right]^{1.5}$$

where

T_c = chimney gas temperature, degrees Centigrade.
μ_o = gas viscosity at O C.
C = constant for specific gas.

Using International Critical Table values, for air $\mu_o = 35.6 \times 10^{-8}$; $C = 124$; for nitrogen $\mu_o = 34.5 \times 10^{-8}$; and $C = 110$.

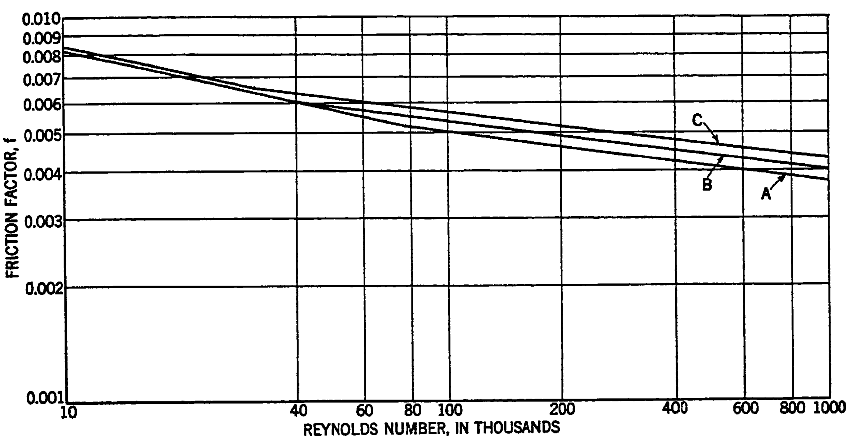


FIG. 5. VARIATION OF FRICTION FACTOR f WITH REYNOLDS NUMBER

Values for the viscosity of air and of nitrogen (the principal component of chimney gases) for the different temperatures follow, in which the values given in pounds-second per square foot are to be multiplied by 10^{-8}:

Temp. F	300	400	500	600	700	800
Air	49.7	54.5	58.5	62.5	66.7	70.5
Nitrogen	47.7	52.2	56.0	59.8	63.5	67.0

Example 2. To determine the Reynolds number C_r for a flow of 118 lb gas per second up a 12 ft diameter chimney at a temperature of 500 F. The gas may be assumed to have the same viscosity as nitrogen at 500 F. Using Equation 5:

$$C_r = 0.0396 \frac{W}{D\mu} = \frac{0.0396 \times 118}{12 \times 56.0 \times 10^{-8}} = 698{,}000$$

The variation of the friction factor f with the Reynolds number is shown in Fig. 5[1]. Three curves are shown: A, B, and C, where the choice of the friction factor curve depends on the relative surface roughness, and this for usual chimney construction may

[1]See also Flow of Fluids in Closed Circuits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

be selected by size since surface conditions in service are always undeterminant. For sizes up to 3 ft in diameter, Curve C may be used; from 3 to 6 ft, Curve B; and from 6 ft upwards, Curve A. Thus for the previous example with $C_r = 698{,}000$ and 12 ft diameter, f would be taken from Curve A as 0.0039.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

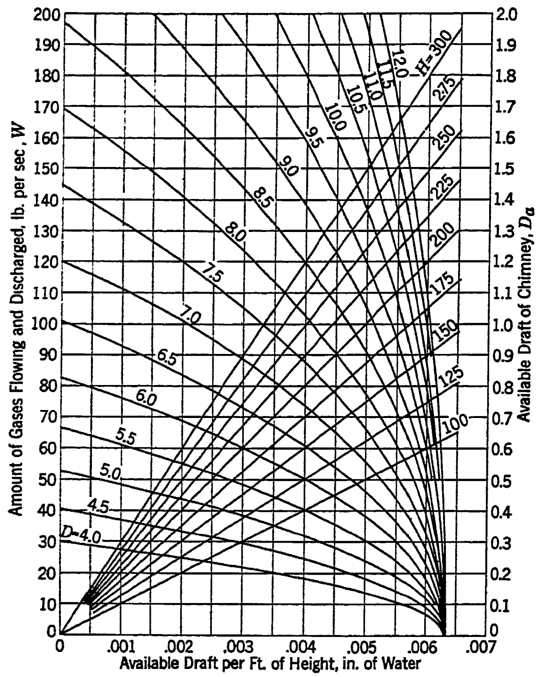


FIG. 6. CHIMNEY PERFORMANCE CHART[a]

[a]*To solve a typical example:* Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

7. Assuming no air infiltration the *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the boiler. The total products of combustion in pounds per second for a grate-fired boiler may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \qquad (6)$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.
G = total grate surface of boilers, square feet.
$C_g \times G$ = total weight of fuel burned per hour.
W_{tp} = total weight of products of combustion per pound of fuel.

A similar computation may be made in the case of gas, oil, or stoker-fired fuel.

Fig. 6 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific operating conditions, a new chart should be constructed from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft chimney are given by the following equations:

$$H = \frac{D_T}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{7}$$

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{8}$$

where

H = required height of chimney above grate bar level, feet.
D = required minimum diameter of chimney, feet (constant for entire height).
V = chimney gas velocity, feet per second.
D_T = total required draft demanded by the entire installation outside of the chimney, inches of water.

Equations 7 and 8 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimneys as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will be least expensive. Since the cost of a chimney structure, regardless

of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi t H D \tag{9}$$

where

Q = volume of material, cubic feet.
t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of HD for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product HD will be least. The solution is obtained by equating the product of Equations 7 and 8 to HD, differentiating this product with respect to V and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{10}$$

where V_e = economical chimney gas velocity, feet per second.

Equation 10 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the corresponding height and diameter can then be determined from Equations 7 and 8, respectively, and the economical size will then be attained. Equations 7, 8 and 10 may be simplified considerably for average operating conditions in an average size steam plant by assuming typical conditions.

Average chimney gas temperature, 500 F.................T_c = 960
Mean atmospheric temperature, 62 F.................T_o = 522
Average coefficient of friction, 0.016.................f = 0.016
Average chimney gas density, 0.09.................W_c = 0.09
Sea level elevation, with barometer of 29.92.................B_o = 29.92

Substituting these values in Equations 10, 8 and 7, respectively, and reducing, the results are substantially:

$$V_e = 13.7 W^{1/5} \tag{11}$$

$$D = 1.5 W^{2/5} \tag{12}$$

$$H = 190 D_r \tag{13}$$

Fig. 7 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 11, 12 and 13. They are based on the operating factors used in reducing Equations 7, 8 and 10 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 7, 8 and 10. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

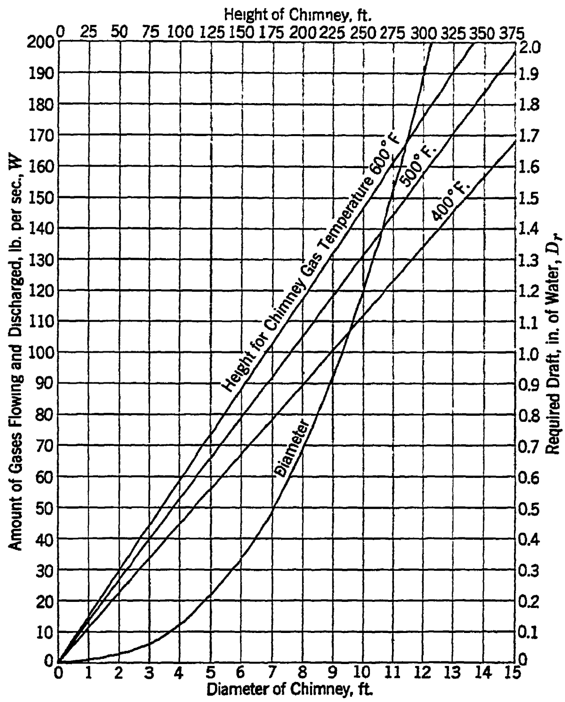


FIG. 7. ECONOMICAL CHIMNEY SIZES[a]

[a]Diameter values also for gas temperatures of 400, 500 and 600 F

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \qquad (14)$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water.
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators, or air heaters, inches of water.

The left hand member of Equation 14 represents the total amount of available draft created by the pressure transformer, that is, the natural draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft intensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \qquad (15)$$

where

D_a = available draft intensity, inches of water.
D_r = required draft, inches of water.

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed, coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 8 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals, much less draft is required than when the particles are small and are well

broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value

TABLE 1. RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES[a]

Warm Air Furnace Capacity in Sq In. of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue		Round Flue		Height in Ft Above Grate
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq In.	Inside Diameter of Lining in Inches	Actual Area Sq In.	
790	590	973	8½x13	7 x 11½	81			35
1000	690	1,140				10	79	
	900	1,490	13x13	11¼x11¼	127			
	900	1,490	8½x18	6¾x16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13x18	11¼x16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18x18	15¾x15¾	248			
	2,480	4,090	20x20	17¼x17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20x24	17x21	357			
	5,000	8,250	24x24	21x21	441			55
	5,570	9,190		24x24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24x28[b]	672			
	8,700	14,400		28x28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30x30[b]	900			
	10,470	17,250		28x32[b]	896			

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[b]Dimensions are for unlined rectangular flues.

should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_B) also varies between wide limits and, in general, depends upon the following factors:

1. Type of boiler.
2. Size of boiler.
3. Rate of operation.
4. Arrangement of tubes.
5. Arrangement of baffles.
6. Type of grate.
7. Design of brickwork setting.
8. Excess air admitted.
9. Location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through

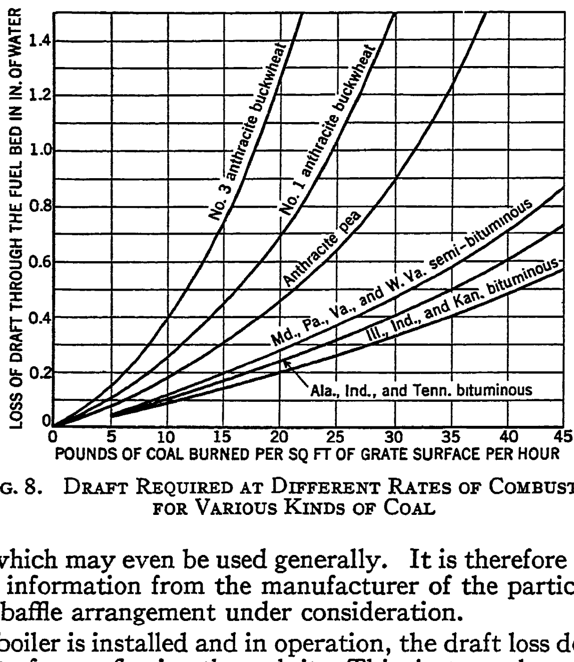


FIG. 8. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. Primarily, the amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being

greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) is given by the general equation:

$$h_{Br} = \frac{0.000194W^2T_cfL}{A^2B_oW_cC_{br}} \tag{16}$$

where

W = the amount of gases flowing, pounds per second.
T_c = absolute temperature of breeching gases, degrees Fahrenheit.
f = coefficient of friction.
L = length of breeching, feet.
A = area of breeching, square feet.
B_o = atmospheric pressure corresponding to altitude, inches of mercury.
W_c = weight of a cubic foot of breeching gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
C_{br} = hydraulic radius of breeching section.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194W^2T_c}{A^2B_oW_c} \tag{17}$$

and represents the amount of draft required to accelerate the gases from zero velocity to the velocity at which the gases are flowing, or in other words, from a static gas condition of zero flow to the amount of gases flowing throughout the installation. This loss corresponds to the velocity head in water works systems.

The *draft loss due to bends* (h_{Bd}) is equivalent to the loss due to the velocity head for a 90-deg bend. In changing direction of flow, the gas velocity decreases to zero with a loss of velocity head and then increases to its proper value at the expense of a loss in pressure head, the net result being a loss in pressure head equal to the velocity head at the bend. This loss is given by the equation:

$$h_{Bd} = \frac{0.000194W^2T_c}{A^2B_oW_c} \tag{18}$$

The friction at a right-angle bend is sometimes expressed as the equivalent of a straight length of flue of a certain length for a certain diameter, similar to the procedure used in estimating the loss due to bends in piping systems conducting water. Most flues, however, particularly breechings, are built square or rectangular in section and no general equation based on the shape of the flue can be conveniently expressed.

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194 K_C W^2 T_C}{A_s^2 B_O W_C} \tag{19}$$

where

K_c = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= 0.5\left(1 - \frac{A_s}{A_1}\right)$

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_O W^2 T_C}{A_s^2 B_O W_C} \tag{20}$$

where

K_o = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= \left(1 - \frac{A_s}{A_1}\right)^2$

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_C}{10^{12}} \tag{21}$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters*. Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The construction, location, height and area of the chimney to which a heating boiler or warm-air furnace is connected affect the operation of the entire heating system. Most residence chimneys are built of brick and may be either lined or unlined, but in either case the *walls must be air-tight* and there should be *only one smoke opening* into the chimney. Cleanout, if provided, must be *absolutely air-tight* when closed.

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining. Fire-clay flue linings shall be manufactured from suitable refractory clay, either natural or compounded, and shall be adapted to withstand high temperatures and the action of flue gases. They shall be of standard commercial thickness, but not less than ¾ in. All fire-clay flue linings shall meet the standard specification of the *Eastern Clay Products Association.* The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining. The wash or splay shall be formed of a rich cement mortar. To improve the draft the wash surface should be concave wherever practical.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped with stone, terra cotta, concrete, cast-iron, or other approved material; but no such cap or coping shall decrease the flue area.

There shall be but one connection to the flue to which the boiler or furnace smoke-pipe is attached. The boiler or furnace smoke-pipe shall be thoroughly grouted into the chimney and shall not project beyond the inner surface of the flue lining.

The size or area of flue lining or of brick flue for warm-air furnaces depends on height of chimney and capacity of heating system. For chimneys not less than 35 ft in height above grate line, the net internal dimensions of lining should be at least 7 x 11½ in.

for a total leader pipe area up to 790 sq in. Above 790 and up to 1,000 sq in. of leader pipe area the lining should be at least 11¼ x 11¼ in. inside. In case of brick flues not less than 35 ft in height with no linings, the internal dimensions should be at least 8 x 12 in. up to 790 sq in. of leader area, and at least 12 x 12 in. for leader capacities up to 1,000 sq in. Chimneys under 35 ft in height are unsatisfactory in operation and hence should be avoided.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

TABLE 2. SUGGESTED GENERAL DIMENSIONS FOR VERTICAL BACK-DRAFT DIVERTER

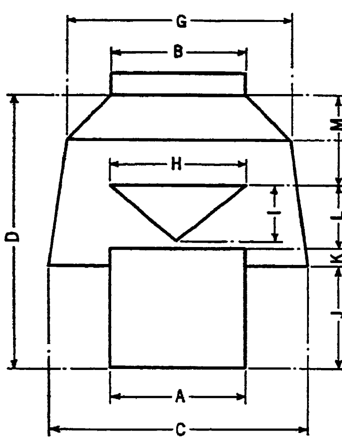


Table of Dimensions (In.)

Pipe Size	A	B	C	D	E	F	G	H	I	J	K	L	M
3	3	3	5.5	7.0	3.8	0.7	4.4	3.0	1.5	2.5	0.7	1.5	2.3
4	4	4	7.2	9.5	5.0	1.0	6.0	4.0	2.0	3.5	1.0	2.0	3.0
5	5	5	9.4	10.8	5.3	1.5	8.0	5.0	2.3	4.0	0.9	2.4	3.5
6	6	6	11.5	12.0	5.6	1.9	9.8	6.0	2.5	4.5	0.8	2.7	4.0
7	7	7	13.5	13.9	6.4	2.3	11.6	7.0	2.9	5.3	0.9	3.1	4.6
8	8	8	15.5	15.8	7.1	2.7	13.4	8.0	3.2	6.0	1.0	3.5	5.3
9	9	9	17.5	17.5	7.7	3.1	15.2	9.0	3.5	6.7	1.0	4.0	5.8
10	10	10	19.7	18.8	7.9	3.6	17.2	10.0	3.8	7.3	1.0	4.3	6.2
11	11	11	22.2	20.7	8.4	4.3	19.6	11.0	4.1	8.0	1.5	4.6	6.6
12	12	12	24.7	22.2	8.7	5.0	22.0	12.0	4.4	8.5	1.7	5.0	7.0

A study of a typical *back-draft diverter* shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value. Converted boilers or furnaces, as well as gas-designed appliances, should be provided with back-draft diverters.

Since back-draft diverters have a special function to perform in protecting gas burning appliances, it is necessary that they should be built to the proper size as shown in Table 2 for a vertical type and in Table 3 for a horizontal arrangement. Equipment of this kind listed by the *American Gas Association* Testing Laboratory must bear the listing symbol *A.G.A.*

TABLE 3. SUGGESTED GENERAL DIMENSIONS FOR HORIZONTAL BACK-DRAFT DIVERTER

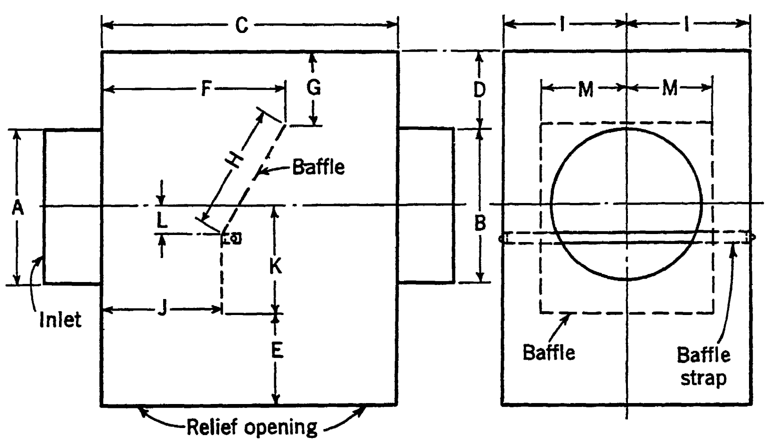


Table of Dimensions (In.)

PIPE SIZE	A	B	C	D	E	F	G	H	I	J	K	L	M
3	3	3	6	1.5	4.8	3.8	1.4	2.5	2.5	2.5	2.1	0 6	1.8
4	4	4	8	2.0	4.8	5.0	1.9	3.4	3.4	3.4	2.9	0.8	2.3
5	5	5	10	2.5	4.8	6.3	2.4	4.2	4.2	4.2	3.5	0.9	2.9
6	6	6	12	3.0	4.8	7.5	2.9	5.0	5.0	5.0	4.3	1.1	3.5
7	7	7	14	3.5	4.8	8.8	3.4	5.9	5.9	5.9	5.0	1.3	4.1
8	8	8	16	4.0	4.8	10.0	3.9	6.7	6.7	6.7	5.6	1.5	4.7
9	9	9	18	4.5	4.8	11.3	4.4	7.5	7.5	7.5	6.4	1.7	5.3
10	10	10	20	5.0	4.8	12.5	4.9	8.4	8.4	8.4	7.0	1.9	5.8
11	11	11	22	5.5	4.8	13.8	5.4	9.2	9.2	9.2	7.8	2.1	6.4
12	12	12	24	6.0	4.8	15.0	5.9	10.0	10.0	10.0	8.5	2.3	7.0

As is the case with the complete combustion of almost all fuels, the products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air into the chimney through the openings in the back-draft diverter, lowers the dew point of the mixture, and reduces the tendency of the water vapor to condense.

The flue connections from a gas-fired boiler or furnace to the chimney should be of a non-corrosive material. In localities where the price of

TABLE 4. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

Height of Chimney Feet	Gas Consumption in Thousands of Btu per Hour								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8.00	9.40	10.50	12.35	13.85
40	4.25	5.50	6.40	7.10	7.80	9.15	10.25	12.10	13.55
60	4.10	5.35	6.20	6.90	7.60	8.90	10.00	11.85	13.25
80	4.00	5.20	6.00	6.70	7.35	8.65	9.75	11.50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

gas requires the use of highly efficient appliances, the material used for the flue connection not only should be resistant to the corrosion of water, but should resist the corrosion of dilute solutions of sulphur trioxide in water. Local practice should be followed in the selection of the most appropriate flue materials.

When condensation in a chimney proves troublesome, it may be necessary to provide a drain to a dry well or sewer. The cause of the excessive condensation should be investigated and remedied if possible. This may be done by raising the flue temperature slightly or increasing the size of the back-draft diverter. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt-chromate provides an excellent protection.

A chimney for a gas-fired boiler or furnace should be constructed in accordance with the principles applicable to other boilers. Where the wall forming a smoke flue is made up of less than an 8-in. thickness of brick, concrete, or stone, a burnt fire-clay flue tile lining should be used. Care should be used that the lengths of flue tile meet properly with no openings at the joints. Cement mortar should be used for the entire chimney.

Table 4 gives the minimum cross-sectional diameters of round chim-

neys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

PROBLEMS IN PRACTICE

1 ● What are the principle factors influencing the intensity of natural draft?

The intensity of natural draft depends largely upon the height of chimney above the grate bar level and the temperature difference between the chimney gases and the atmosphere.

2 ● What two kinds of draft need be considered?

Natural draft caused by temperature differences, and artificial draft caused by mechanical forcing.

3 ● What is the effective height of a chimney?

The height from the grate level to the top of the chimney is the effective height in producing natural draft.

4 ● What dual purpose does a tall chimney fulfill?

A tall chimney primarily creates the necessary draft to move the air required for the combustion process and to move the products of combustion, and secondarily it discharges the gases at a high elevation to prevent them from becoming a nuisance.

5 ● What is the direct influence of the height on the design of a chimney?

The immediate purpose of height is to provide that draft intensity under the conditions of chimney gas temperature such that it will be adequate to overcome all the frictional resistances of the installation, as well as to provide for the actual gas movement.

6 ● Of what importance is chimney cross-sectional area in stack design?

The area should be as large as is economically feasible in order that the frictional loss for the chimney height should not destroy the effectiveness of the height-created draft in overcoming the necessary frictional resistances of the boiler and its flue connections.

7 ● Of what importance is the Reynolds number in chimney design?

It permits the selection of a more specific value of the chimney friction factor, rather than a general one, to correspond with conditions of size, nature of the gas, rate of gas flow, and condition of the surface.

8 ● a. Name the principle advantages of natural draft.
b. Name the principle disadvantages of natural draft.

a. Simplicity, reliability, freedom from mechanical parts, low cost of maintenance, relatively long life, relatively low depreciation, operation with no power requirement.

b. Lack of flexibility, irregularity, dependence on surroundings, susceptibility to temperature changes.

9 ● How is mechanical draft created?

By forced draft, by induced-draft fans, or by a Venturi chimney.

10 ● Distinguish between theoretical and available draft.

Theoretical draft is the difference in pressure inside and outside the base of a chimney when it is under operating temperatures but when there are no gases flowing. Available draft is less than theoretical draft by the friction loss due to the flow of gases through the chimney.

11 ● Explain the term efficiency of a natural draft chimney.

The efficiency of a chimney is the ratio of the work it does in moving gases to the theoretical amount of power it generates.

12 ● How is the available draft used in a heating plant?

The available draft at the base of the chimney is used to overcome the loss in pressure through the grate, the fuel bed, the boiler passes, the breeching.

13 ● What are some of the factors that influence the draft loss through the fuel bed?

Uniformity and size of coal, the amount of ash mixed with the fuel on the grate, thickness of fuel bed, rate of combustion, amount of air supply as related to the coal burning rate.

14 ● How does the volatile matter content affect the draft loss through the fuel bed?

The higher the volatile content and the lower the fixed carbon content, the lower the draft loss.

15 ● In what cases will there be no fuel bed draft loss?

In oil, gas, and powdered fuel firing the fuel is mixed and burned in suspension; consequently, no measurable resistance is encountered in the combustion zone.

16 ● Is it possible to state an average value for the draft loss through a boiler and its setting?

No. The draft loss varies widely and depends on many factors such as the size and type of gas passageways. The manufacturer is usually able to supply such information.

17 ● Of what significance is the CO_2 content of stack gases in establishing draft loss?

The CO_2 content of the exit gases is a measure of the completeness of the combustion and the amount of excess air supplied. Low CO_2 indicates a high excess of air and hence a high draft loss.

18 ● What two effects does an economizer have on the draft loss?

An economizer offers resistance to the flow of gases over the added surfaces; it lowers the temperature of the gases going to the chimney and therefore decreases the available draft. This decrease often necessitates the addition of forced draft.

19 ● What main provisions should be considered in good chimney construction?

Chimneys should be air-tight and connected to only one smoke opening. The chimney top should be high enough above surroundings so the wind will not strike it at any angle above the horizontal. Chimney walls should be not less than one brick in width, and they should be lined with fire-clay tile of the size required for the attached heating unit. Tile lining sizes are stated as outside dimensions; therefore, their effective dimensions are less by the thickness of the wall.

20 ● What is the purpose of a back draft diverter as used on gas burning units?

Since the fuel is supplied under pressure independent of draft it is necessary to free the unit from the variable chimney draft and to supply air for combustion in direct proportion to the supply of fuel gas. The back draft diverter protects the pilot and burners from down drafts.

Chapter 11

AUTOMATIC FUEL BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances

AUTOMATIC mechanical equipment for the combustion of solid, liquid and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

Stokers may be divided into four types according to their construction, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type.

Overfeed Flat Grate Stokers

This type is represented by the various chain- or traveling-grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the stoker to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed. A typical traveling-grate stoker is illustrated in Fig. 1.

Another and distinct type of overfeed flat-grate stoker is the spreader (Fig. 2) or sprinkler type in which coal is distributed either mechanically or by air over the entire grate surface. This type of stoker has a wide application on small sized fuels and on certain special fuels such as lignites, high-ash coals, and coke breeze.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat grate stoker, but this stoker (Fig. 3) is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free

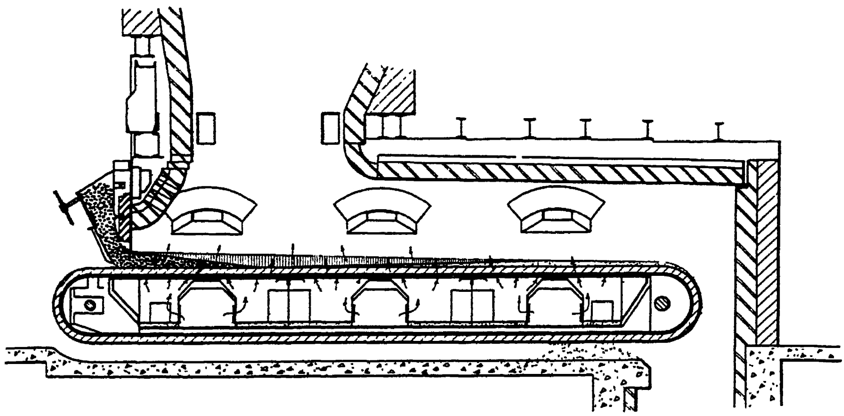


FIG. 1. OVERFEED TRAVELING-GRATE STOKER

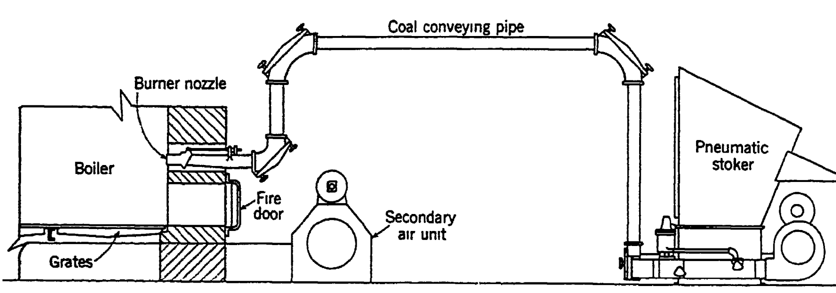


FIG. 2. SPREADER STOKER-PNEUMATIC TYPE

passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gases.

Underfeed Side Cleaning Stokers

In this type (Fig. 4), the fuel is introduced at the front of the furnace to one or more retorts, is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all coking coals while in the smaller sizes it is suitable for small sizes of anthracites. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process

the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously discharged as in the small stoker or may be accumulated on a dump plate and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gases.

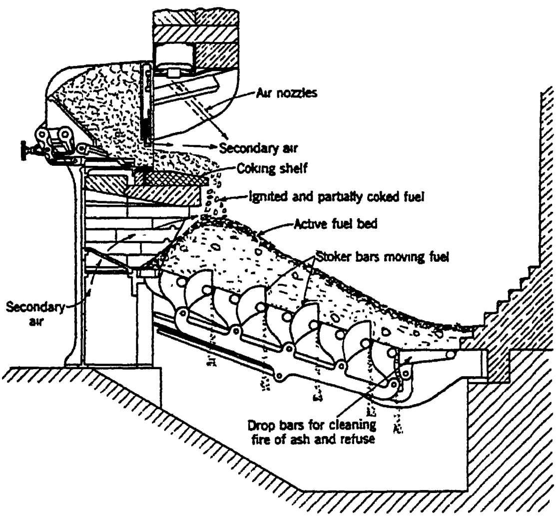


FIG. 3. OVERFEED INCLINED GRATE STOKER

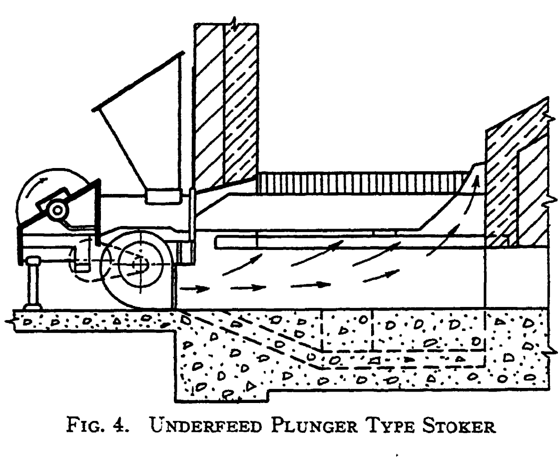


FIG. 4. UNDERFEED PLUNGER TYPE STOKER

Underfeed Rear Cleaning Stokers

This type of stoker accomplishes combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed.

Stokers also may be classified according to their size based upon coal feed rates. The following classification has been made by the *United States Department of Commerce* in cooperation with the *Stoker Manufacturers' Association*.

Class 1. Up to and including 60 lb of coal per hour.

Class 2. 60 to 100 lb of coal per hour.

Class 3. 100 to 300 lb of coal per hour.

Class 4. 300 to 1200 lb of coal per hour.

(A fifth class is included covering stokers having a feeding capacity above 1200 lb of coal per hour).

Class 1 and Class 2 Stokers, Household

Since these stokers are used primarily for home heating, it is desirable that their design be simple and attractive in appearance, and that they be quiet and automatic in operation.

A common type of stoker in this class consists, essentially of a coal reservoir or hopper, a screw for conveying the fuel from the hopper to the burner head or retort, a fan which supplies the air for combustion, a

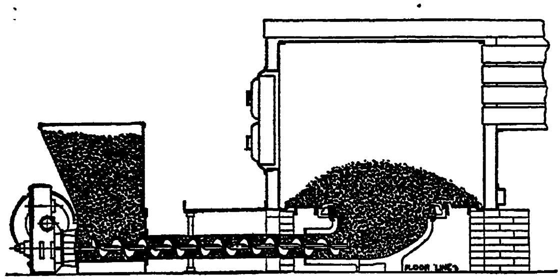


FIG. 5. UNDERFEED SCREW STOKER, HOPPER TYPE

transmission for driving the coal feed worm, and an electric motor or motors for supplying the motive power for both coal feed and air supply as indicated in Fig. 5. The shape of the retort in this class of stokers is usually round although rectangular retorts are favored by some manufacturers. In all cases, however, the retort incorporates tuyeres through which the air for combustion is admitted.

Some household stokers are provided with an automatic grate-shaking mechanism together with screw conveyors for removing the ash from the ashpit (Fig. 6) and depositing it in an ash receptacle outside the boiler.

Certain types can also be provided with a coal conveyor which takes coal from the storage bin and maintains a full hopper at the stoker. In some cases the coal bin functions as the stoker hopper as shown in Fig. 7, and an extended worm is used to convey the fuel to the combustion furnace.

Domestic stokers may feed coal to the furnace either intermittently or with a continuous flow regulated automatically to suit conditions.

Household stokers are made for all classes of fuel; anthracite, bituminous and semi-bituminous coals, and coke. The *United States Depart-*

ment of Commerce has issued commercial standards for household anthracite burners, which may be obtained by application. Standards of performance for bituminous coal stokers are also being developed by *Bituminous Coal Research, Inc.* The standards for anthracite stokers are described in the next paragraphs.

Operating Requirements for Anthracite Stokers

Efficiency. The over-all efficiency of the unit at all points above 50 per cent of maximum coal feed shall be above 50 per cent when installed in a round sectional cast-iron boiler having three intermediate sections and 1½ in. of asbestos insulation or its equiva-

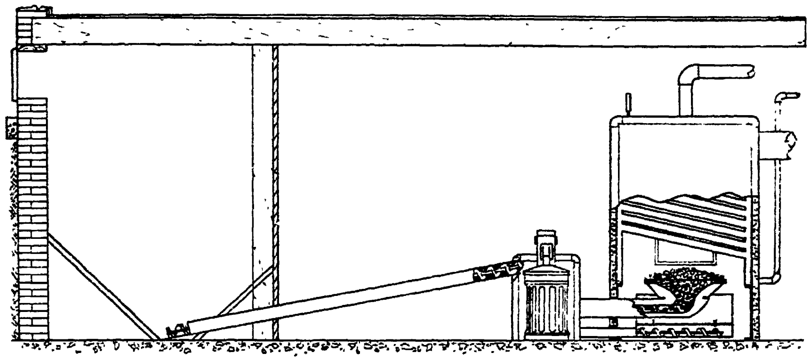


FIG. 6. UNDERFEED SCREW STOKER WITH AUTOMATIC ASH REMOVAL

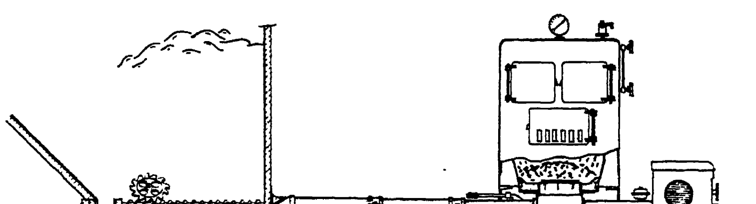


FIG. 7. UNDERFEED SCREW STOKER, BIN FEED TYPE

lent in good condition of repair, operating at 50 per cent or more of the boiler capacity. The efficiency shall be maintained for any continuous period of 4 hours during any test or observation run.

Ash Loss. Combustible in ash shall not exceed 7.5 per cent of the Btu content of the coal as fired at any rate of coal feed above 50 per cent of maximum. Methods of test according to Code No. 3 of the A.S.H.V.E.[1] are to be followed in all details applicable to stoker testing.

Clinker. Ash removing systems should at all times be capable of disposing of any clinker which may be formed under any conditions of operation with the coals prescribed.

[1]A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3), (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 332).

Combustion Rate. A combustion rate of at least 13 lb per square foot of horizontal projected area of ash ring per hour must be continuously maintained for at least 9 hours with the above conditions of efficiency, ash and clinker.

Flue Gas. Flue gas shall be not below 6 per cent in carbon dioxide with a reasonably tight boiler at any rate of operation above 50 per cent of maximum coal feed.

Maximum Rating. The maximum rating, in terms of gross square feet of water or steam radiation which the burner will supply, when intended for installation in the average existing cast-iron boiler, shall be 90 per cent of the maximum steam produced in a round cast-iron boiler in good repair having three intermediate sections and the equivalent of 1½ in. of asbestos insulation. However, in no case shall the maximum rating be greater than 29 sq ft of direct steam radiation for each pound of coal fired per hour, and in no case shall ratings be based upon efficiency figures below 50 per cent.

The maximum rating as defined in the preceding paragraph shall be based upon combustion of Pennsylvania anthracite having the following approximate analysis:

Volatile matter 3.5 to 9 per cent; ash content not to exceed 15 per cent; sulphur content under 1.5 per cent; ash fusing temperature 2750 F, or above (volatile, ash and sulphur content on dry basis in accordance with *A.S.T.M.* method D271-33); Btu content 12,000 or above; properly sized as follows: A No. 1 buckwheat should pass through a round mesh screen having 9⁄16 in. holes and over a similar screen having 5⁄16 in. holes. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent. A No. 2 buckwheat (rice) should pass through a round mesh screen having holes 5⁄16 in. in diameter and over a like screen having holes of 3⁄16 in. in diameter. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent.

Coal Storage. It is recommended that the coal bin or closet be constructed so as to be dustproof.

Electrical Consumption. The electrical consumption shall not exceed 18 kwh per 2000 lb of coal burned at any rate of coal feed above 50 per cent of the maximum.

Operation Upon Other Sizes of Coal. The foregoing specifications have been drafted for operating with the Nos. 1 and 2 buckwheat sizes of anthracite. In the event that other sizes are recommended, ratings shall be based upon the same efficiency and ash loss requirements.

Banking. The burner shall be so constructed or controlled as to maintain a fire during an indefinite banking period.

Acceleration. When the burner resumes operation after a 12 hour banking period, the time required for the stack temperature to reach a normal maximum shall not exceed 60 min.

Stoker-Fired Units

Boilers and air conditioners especially designed for stokers are now becoming available. Present designs feature better coordination between the heat absorber and the stoker. Increased setting height and furnace volume and more heating surface than in conversion installations are usually to be found in these stoker units.

Class 3 Stokers, Apartment House, Small Commercial

This class is used extensively for heating plants in apartments and hotels, and also for small industrial plants such as laundries, bakeries, and creameries. The majority of stokers used in this field are of the underfeed type. The principal exception is an overfeed type having step action grates in a horizontal plane and so arranged that they are alternately moving and stationary, and are designed to advance the fuel during combustion to an ash plate at the rear.

All of the stokers are provided with a coal hopper outside of the boiler. In the underfeed types, the coal feed from this hopper to the furnace may be accomplished by a continuously revolving screw or by an intermittent

plunger. The drive for the coal feed may be an electric motor, or a steam or hydraulic cylinder. With an electric motor, the connection between the driver and the coal feed may be through a variable speed gear train which provides two or more speeds for the coal feed; or it may be through a simple gear train and a variable speed driver for the change in speed of the coal feed; or a simple gear train with a coal feed having an adjustment for varying the travel of the feeding device. With a steam or hydraulic cylinder, the power piston is connected directly to the coal feeding plunger.

The stokers in this class vary also in their retort design according to the fuels and load conditions. The retort is placed approximately in the middle of the furnace and is provided with tuyere openings at the top on all sides. In the plunger-feed type the retort extends from the inside of the front wall entirely to the rear wall or to within a short distance of the rear wall. This type of retort has tuyeres on the sides and at the rear.

These stokers also differ in the grate surface surrounding the retort. In many of the worm-feed stokers this grate is entirely a dead plate on which the fuel rests while combustion is completed. In the dead-plate type, all of the air for combustion is furnished by the tuyeres at the retort. Because of this, combustion is well advanced over the retort so that it may easily be completed by the air which percolates through the fuel bed. With the dead-plate type of grate the ash is removed through the fire doors and it is therefore desirable that the fuel used shall be one in which the ash is readily reduced to a clinker at the furnace temperature, in order that it may be removed with the least disturbance of the fuel bed.

In other stokers in this class, the grates outside of the retort are air-admitting and some stokers have shaking grates. These grates permit a large part of the ash to be shaken into the ashpit beneath, while the clinkers are removed through the fire doors. With this type of grate, the main air chamber extends only under the retort while the side grates receive air by natural draft from the ashpit.

In still other stokers of this class, the main air chamber extends beyond the retort and is covered with fuel-bearing, air-supplying grates. With this type of grate, the fuel is supplied with air from the main air chamber throughout combustion. Also with this type of grate, dump plates are provided beyond the grates where the ash accumulates and from which it can be dropped periodically into the ashpit beneath.

Stokers in this class are compactly built in order that they may fit into standard heating boilers and still leave room for sufficient combustion space above the grates. The height of the grate is approximately the same as that of the ordinary grates of boilers, so that it is usually possible to install such stokers with but minor changes in the existing equipment. In some districts, there are statutory regulations governing such settings.

These stokers vary in furnace dimensions from 30 in. square to approximately 66 in. square. The capacity of the stokers is measured by the amount of coal that can be burned per hour. In general, manufacturers recommend that, for continuous operation, the coal burning rate shall not exceed 25 lb of coal per square foot of grate per hour, while for short peaks this rate may be increased to 30 lb per hour. Although these stokers were designed to burn bituminous coal, types are available for

the semi-bituminous coals such as Pocahontas and New River. They can also be used to burn the small sizes of anthracite but at a somewhat lower rate.

Class 4 Stokers, Medium Commerial

These stokers are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. Rectangular retorts with sectional tuyeres and dead plates without air ports are employed. The unit type of construction is almost universally used, the unit incorporating the hopper, the transmission for driving the feed screw, and the fan for supplying air for combustion.

Class 4 Stokers, Large Commercial, Small High Pressure Plants

Stokers in this group vary widely in details of mechanical design and the several methods of feeding coal previously described may be employed Such methods of applying power to the fuel conveying mechanism as, continuous gear train transmission, ratchet type speed reducer, hydraulic cylinder and steam cylinder are used. Varying methods of ash disposal are found in this class.

Large Stokers

This class includes stokers with hourly burning rates of over 1200 lb of coal per hour. The prevalent stokers in this field are:

a. Overfeed flat grate stokers.
b. Overfeed inclined grate stokers.
c. Underfeed side cleaning stokers.
d. Underfeed rear cleaning stokers.

Overfeed inclined grate stokers are seldom built in sizes of over 500 hp and are not as extensively used as other types of stokers.

Underfeed side cleaning stokers are made in sizes up to approximately 500 hp and in this field are extensively used. These stokers are not so varied in design as those in the smaller classes although the principle is much the same. Practically all of them are of the front coal feed type, either power driven or steam driven. Dump plates at the side are manually operated. These stokers are heavily built and designed to operate continuously at high boiler ratings with a minimum amount of attention. Because of the fact that all volatile gases must pass through the fire before reaching the combustion chamber, these stokers will operate smokelessly under ordinary conditions. Also because of the fact that these stokers are always provided with forced draft, they are the most desirable type for fluctuating loads or high boiler ratings.

In the design of the grates for supporting the fuel between the retort and the ash plates, the stokers differ in providing for movement of the fuel during combustion. Some stokers are designed with fixed grates of sufficient angle to provide for this movement as the bed is agitated by the incoming fuel, while others have alternate moving and stationary bars in this area and provide for this movement mechanically. In either type, with proper operation, all refuse will be deposited at the dump plate.

Recent developments in this type of stoker provide for sliding distributor blocks along the bottom of the retorts which give flexibility in providing proper distribution of fuel over the grate area and assist in preventing coke masses when strong coking coals are used. Another difference in these stokers is that some use a single air chamber under the whole grate area thus having the same air pressure under the ignition area as under the rest of the grate, while others have a divided air chamber using the full air pressure under the ignition area and a reduced air pressure under the remainder of the grate. These stokers vary in size from approximately 5 sq ft to a maximum of 8½ sq ft.

The most prevalent type of rear cleaning underfeed stoker is the multiple retort design. Occasionally double or triple retort side cleaning underfeeds are made. The multiple retort underfeed stoker is made for the largest sizes of boilers for large industrial plants and central stations. This stoker has reached a very fine stage of development mechanically and in the matter of air supply and control. In some instances zoned air control has been applied both longitudinally and transversely to the grate surface. Ash dumps on smaller sizes are sometimes manually operated.

The Combustion Process

Due to the marked differences in design and operating characteristics of stokers and the widely differing characteristics of stoker fuels, it is difficult to generalize on the subject of combustion in automatic stokers.

In anthracite stokers, which are almost exclusively of the small (Class 1) underfeed type, burning takes place within the stoker retort. The ash and refuse of combustion spills over the edge of the retort into an ashpit or receptacle from which it may be removed either manually or automatically. Anthracite is usually supplied for stoker firing in No. 1 buckwheat or No. 2 buckwheat size. Those stokers burning coke operate in a similar manner to anthracite stokers.

Since the majority of bituminous coal stokers used in heating plants operate on the underfeed principle some general observations of their operation are given.

When the coal is fed from the hopper or bin into the retort it is generally degraded to some extent and some segregation of sizes occurs. Because of these factors there may be some difference in the actions occurring in the various portions of the retort.

The coal moving upward in the retort toward the zone of combustion established by previous kindling of the fire is heated by conduction and radiation from the zone of combustion. As the temperature of the coal rises it first gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to around 700 or 800 F, the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of combustible volatile matter occurs during and directly after the plastic stage of the coal. The distillation of volatile matter continues above the plastic zone and the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal used.

As more coal is fed from below the mass of coke continues to grow forming a coke *tree*, *plug* or *spar* as it is variously designated. After a period of time, dependent upon the strength of the coke formed, pieces of the coke *tree* break off and fall upon the hearth surrounding the retort or within the retort itself where they are burned.

While part of the ash fuses into particles at the surface of the coke as it is released, most of it is freed in unfused flakes or grains. The greater part of this unfused ash remains on the hearth or dead plates although a part may be expelled from the furnace with the gases.

The ash layer becomes thicker with time and that near the retort, being exposed to temperatures which are high enough at times, fuses into a clinker. The temperature attained in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating are factors which govern the degree of fusion.

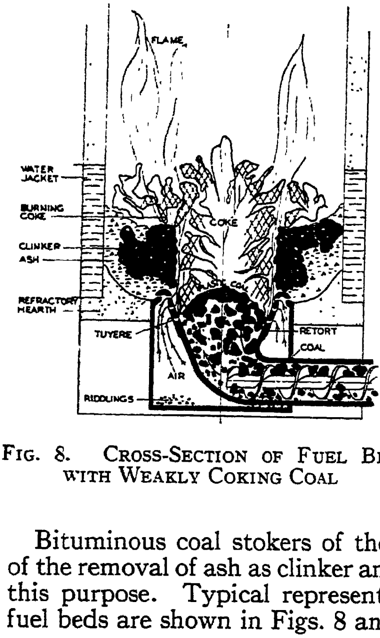


FIG. 8. CROSS-SECTION OF FUEL BED WITH WEAKLY COKING COAL

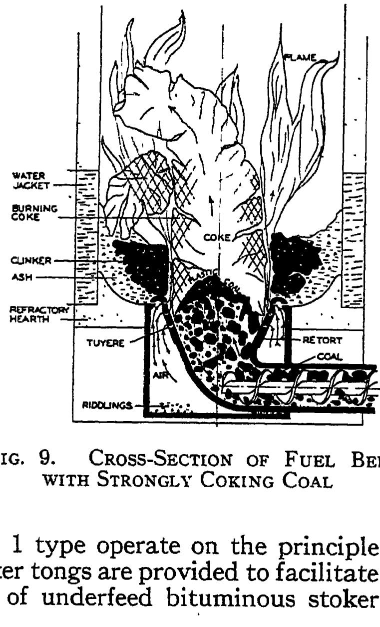


FIG. 9. CROSS-SECTION OF FUEL BED WITH STRONGLY COKING COAL

Bituminous coal stokers of the Class 1 type operate on the principle of the removal of ash as clinker and clinker tongs are provided to facilitate this purpose. Typical representations of underfeed bituminous stoker fuel beds are shown in Figs. 8 and 9.

The appearance of such fuel beds is very ragged at times, and large masses of coke build up, surrounded by blowholes with intense white flame indicating the presence of excess air. There is a natural tendency for users to disturb the fuel bed and make it conform to the conventional representation or ideal fuel bed. Such attention should not be required, as usually the fuel bed tends to correct its own faults as the cycles of plasticity, coke *tree* formation and ash fusion recur.

There are a number of factors which materially affect the rate and type of combustion obtained in stoker usage, the most important of these being: the type and design of stoker, the type and characteristics of the fuel, the method of stoker installation and the method of stoker operation.

Furnace Design

The burning of the fuel on the grate or in the retort will be influenced directly by the stoker design. The burning of the volatile gases above the fuel bed is a matter of furnace design. Proper care should be taken to provide furnaces sufficiently liberal in volume and with the grates or retorts at a sufficient distance from the heating surface to permit proper combustion of the gases. Smoke and low efficiency will result if the furnace is too small to permit proper mixing of the gases and completion of combustion.

TABLE 1. RECOMMENDED SETTING HEIGHTS FOR HEATING BOILERS EQUIPPED WITH MECHANICAL STOKERS[a]

FIREBOX BOILERS

Actual Load[b]	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18″	18″	20″	20″	22″	22″	24″	24″	24″
B	42″	48″	54″	60″	66″	72″	78″	84″	84″

A = Distance from bottom of Water Leg to floor. *B* = Distance from Crown Sheet to bottom of Water Leg.

COMPACT WELDED BOILERS

Actual Load[b]	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18″	18″	20″	20″	22″	22″	24″	24″	24″
B	30″	33″	36″	42″	45″	48″	54″	60″	60″

A = Distance from bottom of Water Leg to floor. *B* = Distance from Crown Sheet to bottom of Water Leg.

H. R. T. BOILERS

Hp	50	75	100	125	150	175	200	225	250	275	300
A	5′-0″	5′-6″	6′-0″	6′-6″	7′-0″	7′-0″	7′-6″	8′-0″	8′-6″	9′-0″	9′-0″

A = Distance from bottom of shell to floor. *Hp* = Installed horsepower.

In the case of the Firebox or Compact Welded type boilers the desired setting height can be obtained by combining *A* and *B* dimensions. The load ratings shown for this class of boilers are actual developed loads in square feet of equivalent cast-iron steam radiation and are not manufacturers' ratings.

The setting heights given for H. R. T. boilers may be used for developed loads up to 50 per cent above normal rating.

[a]From Data prepared by the *Midwest Stoker Association*.

[b]Expressed as steam radiation, 1 sq ft = 240 Btu.

The standards that have been most universally adopted for the proportioning of furnaces for bituminous coal stokers are those of the *Midwest Stoker Association* and the *Steel Heating Boiler Institute*. See Chapter 13.

Table 1 gives recommended setting heights for heating boilers equipped with mechanical stokers using bituminous coal.

Furnace volume is not an important item in anthracite stoker installations. Due care should be exercised for both anthracite and bituminous stokers to prevent intense heat application on the metal surfaces of the combustion chamber. The installation of a baffle or adjustment in setting height of the stoker may be desirable in some cases.

The prime essentials of good furnace design are: correct proportions, moderate combustion rate, adequate furnace volume and sufficient flame clearance. If these factors are properly compensated for and provision is made for the proper mixing of the gases bituminous coal stokers will operate smokelessly. In those stokers which are operated intermittently, however, some smoke may be produced during the *off* periods.

Combustion Adjustments

Satisfactory stoker performance may be secured by regulating the coal feed and the air supply so as to maintain, as nearly as possible, an ideal balance between the load demand and the heat liberated by the fuel. When the coal is consumed at about the same rate as that which it is fed this balance exists and uniform fuel bed conditions will be found. Under such conditions no manual attention to the fuel bed should be required other than the removal of clinker in those stokers which operate on this principle of ash removal.

Since complete combustion is not obtained in stoker furnaces receiving only the air theoretically required, it is necessary, even under the best of conditions, to supply from 30 to 50 per cent excess air to obtain desired combustion results. Due to the variable characteristics of solid fuels in burning, consideration must be given to a number of factors which affect the maintenance of the combustion conditions wanted.

The specified rate of coal feed of a stoker may vary due to changes in the bulk density of the coal dependent upon: (*a*) the size of coal, (*b*) distribution of size in the coal, (*c*) segregation of coal in the stoker hopper, and (*d*) friability of the coal.

The following factors may affect the rate of air supply: (*a*) changes in fuel bed conditions and resistance, (*b*) changes in furnace draft due to a variety of causes, i.e., changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues, and (*c*) changes in air inlet adjustments to the fan.

Many domestic bituminous stokers now incorporate some method of automatic control which compensates for changes in fuel bed resistance. Since a secondary source of air due to leakage is present in most installations, the use of an automatic draft regulator to maintain the furnace draft at about 0.05 in. of water is desirable. This is quite important with intermittently operated stokers. Some fuel is burned by natural draft in the *off* periods, when fuel is not being fed, and it is essential that the burning in these periods be controlled. With excessive draft, due either to fan pressure or chimney pull, an increase in the discharge of soot and fly ash from the combustion chamber will result.

Measurement of the Efficiency of Combustion

As efficient combustion is based upon a certain percentage of excess air, it is possible to determine the results by analysis of the gases formed by the combustion process. An Orsat apparatus can be used to determine the percentage (by volume) of the carbon dioxide (CO_2), oxygen (O_2) and

carbon monoxide (*CO*) in the flue gases. Due to variations in the fuel bed and rate of burning of stoker-fired solid fuels it is not sufficient to analyze a *grab* sample. A continuous gas sample drawn at a constant rate throughout an operating period of reasonably long duration should be used.

A CO_2 reading of 12.5 to 14 per cent indicates that the excess air supplied is in the range of 30 to 50 per cent. The presence of *CO* indicates a loss due to improper mixing of the air and the gases of combustion. As an increase in excess air maintained during *on* periods will decrease the tendency toward smoke in the *off* periods of intermittently operated bituminous coal stokers, care should be taken that the delivery of air by the fan is great enough to avoid smoke.

Controls

The industry developed by stokers in Classes 1, 2, 3 and 4 has been due as much to the application of proper controls as to the stoker itself. This is especially true of Class 1 stokers because of their application to residential heating, a field wherein the majority of owners and users are not familiar with control problems or stoker operation.

The usual controls applied are as follows:

a. Thermostats (plain and clock).
b. Limit Controls (steam, vapor, vacuum, hot water, or air).
c. Stack Temperature or Time Controls (for actuating fires periodically).
d. Relay (for low voltage controls).
e. Safety or Overload Cutout (for protection against overload).
f. Low Water Cutout (steam, vapor and vacuum).

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically and safely from liquid fuels. This heat is produced in the furnace or firepot of hot water or steam boilers or warm air furnaces and is absorbed by the boiler, and thus made available for distribution to the house through the heating system.

With oil, as with any kind of fuel, efficient heat production requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with a minimum of excess air. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil, composed of carbon and hydrogen, will be completely oxidized and thus produce all the heat possible. The use of unreasonable quantities of air in excess of theoretical combustion requirements results in lowered efficiencies due to increased stack losses. Such losses, if not accompanied by unburned products of combustion (saturated and unsaturated hydrocarbons, hydrogen, etc.), may be offset somewhat by increasing the secondary heating surfaces of the heat absorbing medium, boiler or furnace.

Oil is a highly concentrated fuel composed mainly of hydrogen and carbon. In its liquid form oil cannot burn. It must be converted into a gas or vapor by some means. If the excess air is to be kept within efficient limits it means that air must be supplied in carefully regulated

quantities. The air and oil vapor must be vigorously mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air that will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will not be stopped before completion. When equipped with a means of igniting the oil and safety devices to guard against mishaps, the oil burner possesses all of the elements to be efficient and automatic.

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION
   - a. *Atmospheric*—by natural chimney draft.
   - b. *Mechanical*—electric-motor-driven fan or blower.
   - c. *Combination of* (*a*) *and* (*b*)—primary air supply by fan or blower and secondary air supply by natural chimney draft.

2. METHOD OF OIL PREPARATION
   - a. *Vaporizing*—oil distills on hot surface or in hot cracking chamber.
   - b. *Atomizing*—oil broken up into minute globules.
     - (1) Centrifugal—by means of rotating cup or disc.
     - (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
     - (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
     - (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
   - c. *Combination of* (*a*) *and* (*b*).

3. TYPE OF FLAME
   - a. *Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
   - b. *Non-luminous*—Bunsen-type flame (*i.e.*, blue flame).

4. METHODS OF IGNITION
   - a. *Electric.*
     - (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating or just at the beginning of operation.
     - (2) Resistance—by means of hot wires or plates.
   - b. *Gas.*
     - (1) Continuous—pilot light of constant size.
     - (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.
   - c. *Combination*—electric sparks light the gas and the gas flame ignites the oil.
   - d. *Manual*—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION
   - a. *On and off*—burner operates only a portion of the time (intermittent).
   - b. *High and low*—burner operates continuously but varies from a high to a low flame.
   - c. *Graduated*—busner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

A trade classification of oil burners consists of the following general types: (*a*) gun or pressure atomizing, (*b*) rotary and (*c*) pot or vaporizing.

The gun type, illustrated in Fig. 10 is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes are employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

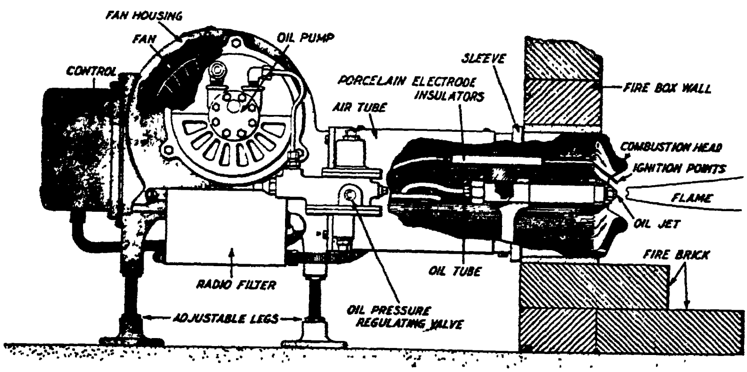


FIG. 10. GUN TYPE PRESSURE ATOMIZING OIL BURNER

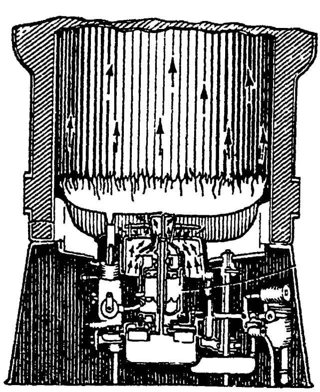


FIG. 11. CENTER FLAME VERTICAL ROTARY BURNER

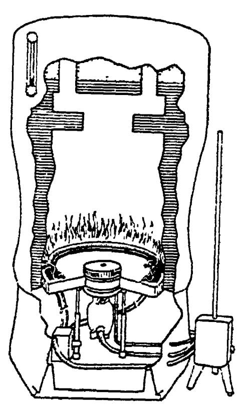


FIG. 12. WALL FLAME VERTICAL ROTARY BURNER

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types; the center flame and wall flame. In the former type, (Fig. 11) the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 12) differs in that combustion takes place in a ring of

refractory material, which is placed around the hearth. These types of burners are further characterized by their installation within the ashpit of the boiler or furnace.

The pot type burner (Fig. 13) can be identified by the presence of a metal structure, called a pot or retort, in which combustion takes place.

When gun type (pressure atomizing) or horizontal rotary burners are used the combustion chamber is usually constructed of firebrick or other suitable refractory material, and is part of the installation procedure.

The oil burners are operated by a small electric motor which pumps the oil and some or all of the air required. The smallest sizes can generally burn not much less than 1 gal of oil per hour. The grade of oil burned ranges from No. 1 to No. 4. No. 4 oil is the heaviest and most viscous of the various grades mentioned. An oil burner satisfactory for No. 4 oil can burn any of the lighter grades easily but an oil burner recommended for No. 2 oil should never be supplied with the heavier grades. It has been found that while the heavier grades of oil have a smaller heat value per

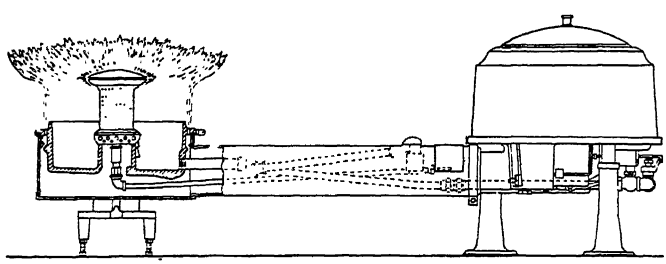


FIG. 13. POT TYPE VAPORIZING BURNER

pound, they have, due to greater density, a larger heat value per gallon. The relative economy of the various grades must be based upon price and the amount of excess air required for clean and efficient combustion.

Boiler-Burner Units

Boilers and air conditioners especially designed for oil burners are available to the purchaser of this type of equipment. They are used for replacements as well as for new installations. This type of equipment usually has more heating surface than the older coal-burning designs. Flue proportions and gas travel have been changed with beneficial results. All problems of combustion chamber design, capacities, efficiencies, etc., have been solved. The selection of the proper size of unit should be a simple process.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat pro-

duction usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups producing a horizontal torch-like flame. (Fig. 14). As much

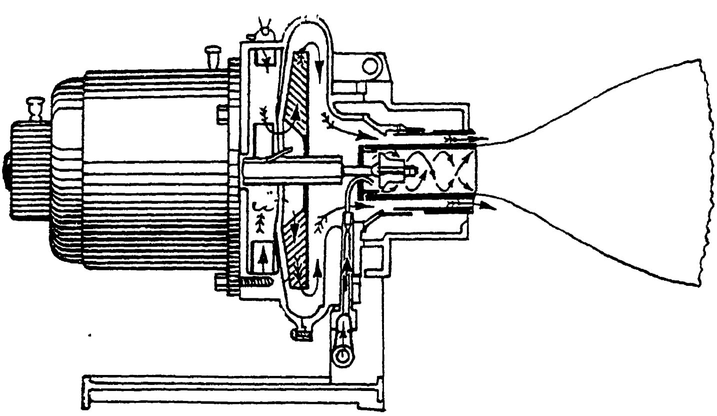


FIG. 14. HORIZONTAL ROTATING-CUP OIL BURNER

as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler, and heating the oil to within 30 deg of its flash point.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil

and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ashpit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2 lb of oil can properly be burned. This corresponds to a maximum liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. This in turn suggests that the value of 38,000 Btu per cubic foot per hour might be adjusted according to good engineering judgment. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

The Combustion Process

Efficient combustion, as previously indicated, must produce a clean flame and must use relatively small excess of air, i.e., between 25 and 50 per cent. This can be done only by vaporizing the oil quickly, completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the liquid fuel to the gaseous state through the application of heat. This is accomplished before the oil vapor mixes with air to any extent and if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced. There may be a deficiency of air as shown by the presence of carbon monoxide (*CO*) or an excessive supply of air, depending upon burner adjustment, without altering the clean, blue appearance of the flame.

An atomizing burner i.e., gun and rotary types is so named because the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result of such practice is the ability to burn more and heavier oil within a given combustion space or furnace volume. Since the air enters the fire pot with the liquid fuel particles, it follows that mixing, vaporization and burning are

all occurring at once in the same space. This produces a luminous instead of a blue or non-luminous flame. In this case a deficient amount of air is indicated by a dull red or dark orange flame with smoky flame tips.

An excessive supply of air may produce a brilliant white flame in some cases or, in others, a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be easily detected, it is generally not possible to distinguish, by the eye alone, the finer adjustment which competent installation requires.

Certain tests indicate that there is no difference in economy between a blue flame and a luminous flame if the position, shape and the per cent of excess air of both flames are about the same.

Furnace or Combustion Chamber Design

The furnace or combustion chamber may be defined as that part of a boiler or conditioner in which combustion is established. With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

It is evident that the atomizing burner is dependent upon the surrounding heated refractory or fire brick surfaces to vaporize the oil and support combustion. While the importance of the combustion chamber is obvious, its design has been troublesome. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the fire brick surface, a carbon deposit will result. Fundamentally, the combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber thus resembles in shape the outline of the flame. In this way as much fire brick as possible is close to the flame so it may be kept quite hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead or inactive spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. With some of the vertical rotary burners considerable care must be exercised in definitely following the manufacturers instructions when installing the hearth as in this class successful performance depends upon this factor.

Combustion Adjustments

Where adjustments of oil and air have been made which give efficient combustion, the problem of maintaining the adjustments constant becomes an important one. Particularly is this true when the change causes the per cent of excess air to decrease below allowable limits of the

burner. A decrease in air supply while the oil delivery remains constant or an increase in oil delivery while the air supply remains constant will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment (*i.e.*, 25 per cent excess air) the more critical it will be of variations. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (*a*) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (*b*) erosion of atomizing nozzle, (*c*) fluctuations in by-pass relief pressures and (*d*) possible variations in methods 2*b* (3) and 2*b* (4) listed in the previous classification table. Note that any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service but usually no soot will form.

The following factors may influence the air supply: (*a*) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues), and (*b*) changes in air inlet adjustments to the fan.

It is recognized that a secondary source of air due to leakage in the boiler setting is present in many installations and it is highly desirable that this leakage be reduced to a minimum. Obviously the amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02-0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Measurement of the Efficiency of Combustion

Efficient combustion being based upon a clean flame and certain proportions of oil and air employed, it is possible to determine the results by analyzing the gases formed by the combustion process. An Orsat apparatus is a device which measures the volume of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. Except in the case of a non-luminous flame it is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations at the present time show from 8 to 10 per cent CO_2. Taking into account the potential hazard of oil or air fluctuations with low excess air (high CO_2) a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners. This is particularly true of non-luminous flame burners which will not function properly with less than 10 per cent CO_2.

Controls

Oil burner controls may be divided into two parts: (*a*) devices to regulate burner operation so the desired house heating result may be obtained and (*b*) devices for the safety and protection of the boiler and burner. For control devices generally consult Chapter 37. The room thermostat has recently been improved to provide more frequent burner

operation and greater uniformity of room temperature. Class (*b*) controls comprises a device to shut off the burner if the oil fails to ignite or if the flame should cease due to lack of oil; a device actuated by steam boiler pressure to shut off the burner when the pressure reaches some predetermined value; a device on the boiler to shut off the burner if the water level acts too low for safety or one which automatically feeds additional water to the boiler; a device on warm air furnaces to shut off the burner if the air temperature gets too high; a valve in the oil supply line which automatically closes in the event of fire in or near the cellar; and a device to keep the temperature of the boiler water within certain limits when it is being used to heat domestic hot water. These devices are all tested and approved by the Underwriters' Laboratory of Chicago, Ill., before they are offered to the purchaser. The selection of class (*b*) devices is made by the oil burner manufacturer.

Fuel Oil Gages

To insure a constant supply of fuel oil and to check deliveries and consumption it is essential to have accurate means of readily determining the quantity of oil in the storage tank. For this purpose various types of indicating or recording gages are used, the simplest forms being the glass level gage, and a float-and-dial arrangement having a graduated dial face indicating the proportion of the tank containing liquid. Other more accurate and dependable devices are designed to operate by hydraulic action or by hydrostatic impulse. These instruments may be attached to the tank, giving a direct reading of the liquid contents; or the instrument itself may be located at a convenient point remote from the tank and connected with the tank by pipe or tubing. The quantity readings may be in gallons of liquid, height of liquid level in feet and inches, or in other desired units of measurement.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

I. Gas-Designed Heating Systems.
   A. Central Heating Plants.
      1. Steam, hot water, and vapor boilers.
      2. Warm air furnaces.
   B. Unit Heating Systems.
      1. Warm air floor furnaces.
      2. Industrial unit heaters.
      3. Space heaters.
      4. Garage heaters.

II. Conversion Heating Systems.
   A. Central Heating Plants.
      1. Steam, hot water and vapor boilers.
      2. Warm air basement furnaces.

These systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push-button control, or manual control.

Gas-Fired Boilers

Information on gas-fired boilers will be found in Chapter 13. Either snap action or throttling control is available for gas boiler operation. This is especially advantageous in straight steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Gas-Fired Warm Air Furnaces

Warm air furnaces are variously constructed of cast iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar as far as the period of heating and temperature to be maintained are concerned.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating and Air Conditioning Association* are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coordinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Floor Furnaces

Warm air floor furnaces are well adapted for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being

suspended below. Air is taken downward between the two sheets of the double casing and discharged upward over the heating surfaces and into the room. The appliance is controlled from the room to be heated by means of a control lever located near the edge of the register. The handle of the control is removable as a precaution against accidental turning on or off of the gas to the furnace.

Space Heaters

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although they should be connected with solid piping it is sometimes desirable to connect them with flexible gas tubing in which case a gas shut-off on the heater is not permitted, and only *A.G.A.* approved tubing should be used.

Parlor furnaces or *circulators* are usually constructed to resemble a cabinet radio. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass up around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters make admirable auxiliary heating appliances to be used during the occasional cool days at the beginning and end of the heating season when heat is desired in some particular room for an hour or two. The radiant heater gives off a considerable portion of its heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire-clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to the ordinary wall register. Others are encased in frames which fit into fireplaces.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator connected to a basement boiler. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to the steam or hot water radiators. They are usually constructed of pressed steel or sheet metal

hollow sections. The hot products of combustion circulate through the sections and are discharged out a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type, operating without refractories.

Many conversion units are equipped with sheet metal secondary air ducts which are inserted through the ashpit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. By means of this duct the air necessary for proper combustion is supplied directly to the burner, thereby making it possible to reduce the amount of excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push-button, or room temperature control.

The Combustion Process

Because of the varying composition of gases used for domestic heating it is difficult to generalize on the subject of gas burner combustion. Refer to the section on Gas Classification, in Chapter 9.

Combustion Adjustments

Little difficulty should be experienced in maintaining efficient combustion conditions when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and, therefore, the rate of heat supply is nominally constant. Since the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft diverter insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft diverter is also helpful in controlling the amount of excess air and preventing back drafts which might extinguish the flame.

Measurement of the Efficiency of Combustion

It is possible to determine the results of combustion by analyzing the gases of combustion with an Orsat apparatus. It is desirable to determine

the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. While ultimate CO_2 values of 10 to 12 per cent may be obtained from the combustion of gases commonly used for domestic heating, a combustion adjustment which will show from 8 to 10 per cent CO_2 represents a practical value. Under normal conditions no CO will be produced by a gas-fired boiler or furnace. Limitations as to output rating by the *A.G.A.* are based upon operation with not more than 0.04 per cent CO in the products of combustion. This is too small an amount to be determined by the ordinary flue gas analyzer.

Controls

Gas burner controls may be divided into two parts: (*a*) devices to regulate burner operation so that the desired house heating results may be obtained and (*b*) devices for the safety of the boiler and burner. Control devices are treated in detail in Chapter 37. A room thermostat may be used as a control of house heating effect. These may be obtained in a number of types. Some central heating plants are equipped with push-button or manual control. Class (*b*) controls include a device to shut off the burner if the gas fails to ignite, a device actuated by boiler pressure, water temperature, or furnace bonnet temperature to shut off the burner when the pressure or temperature becomes excessive, a device on the boiler to shut off the burner if the water level falls below safe limits or one which automatically feeds additional water to the boiler, and a device for controlling the gas pressure within desired limits. The main gas valve may be either of the snap action or throttling type.

Sizing Gas-Fired Heating Plants

While gas-burning equipment can be and usually is so installed as to be completely automatic, maintaining the temperature of rooms at a pre-determined and set figure, there are in use installations which are manually controlled. Experience has shown that, in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a much lower selection, or safety factor. A gas-fired boiler under thermostatic control is sensitive to variations in room temperatures so that in most cases a factor of 20 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Consequently a selection factor for thermostatically controlled boilers must be variable. Liberal selection factors to be added to the installed steam radiation under thermostatic control are given in Fig. 3 of Chapter 13.

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 foı steam, and 150 for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. This same limitation applies to all classes of gas-consuming heating appliances that are tested and approved by the Laboratory. Gas boilers are available with ratings up to 14,000 sq ft of steam, while furnaces with ratings up to about 500,000 Btu per hour are available. (See Chapter 20.)

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PROBLEMS IN PRACTICE

1 ● List some factors which might account for higher efficiencies with stoker firing than with hand firing.

a. The uniform rate of coal feed, *b.* Better distribution in the fuel bed, and *c.* Positive control of the air supplied for combustion.

2 ● Classify stokers as to construction and operation.

a. Overfeed flat grate, *b.* Overfeed inclined grate, *c.* Underfeed side cleaning type, and *d.* Underfeed rear cleaning type.

3 ● What classification may be made of stokers as to their use?

Class 1. For residences (Capacity up to 60 lb of coal per hour).

Class 2. For apartment houses and small commercial heating plants (Capacity 60 to 10C lb of coal per hour).

Class 3. For medium sized commercial heating plants (Capacity 100 to 300 lb of coa per hour).

Class 4. For large commercial plants and small high pressure steam plants (Capacit; 300 to 1200 lb of coal per hour).

4 ● What main parts are found in an underfeed residential stoker?

A ***hopper*** is supplied to hold coal which is fed by a ***screw*** or ***plunger*** into a ***retort*** provide with air openings called ***tuyeres***. A ***blower*** supplies air under pressure for combustion, an a ***gear case*** provides for changes in coal feeding rates.

5 ● What is a dead-plate?

A dead-plate is a flat surface without air supply openings upon which the fuel rests whi combustion of the fixed carbon is completed. Generally the ash is removed from tl dead-plate.

6 ● What features of furnace design are essential for the proper burning of the volatile coal gases above the fuel bed?

Adequate provisions should be made so that the furnace volume is sufficiently liberal and that the grates are a sufficient distance from the heating surfaces to permit the proper combustion of gases.

7 ● What methods of oil atomization are used?

a. Throwing the oil from a rotating cup or disc, *b*. Forcing the oil under high pressure through a nozzle, *c*. Propelling the oil with a high velocity jet of air or steam, and *d*. Forcing an oil and air mixture through a nozzle.

8 ● What is the purpose of atomization?

Atomization is used to increase the surface area of the oil in order to facilitate putting it into a vaporous state so it may burn.

9 ● Is the furnace of much importance in oil burning?

In most cases it is very important. It is the function of the oil burner to supply the air and fuel in correct proportions; the furnace must provide heated space for proper mixing and combustion.

10 ● Which flame is considered better, the luminous or the non-luminous?

Laboratory tests show that they are equally efficient in the usual installation.

11 ● How should oil burner adjustments be made?

Adjustments should be made by an experienced man who uses a gas analysis apparatus to determine the CO_2 content.

12 ● What CO_2 content should be attained in oil burning?

Ten per cent CO_2 is considered good practice, for it indicates the supplying of 50 per cent excess air.

13 ● What maximum heat release is considered good practice in oil burning?

A heat release of 38,000 Btu per cubic foot per hour is considered to be the maximum for average large installations. This figure has been greatly exceeded in some cases. The design of the combustion chamber, as to impingement of flame and as to proper mixing at high temperatures, has much to do with the attainable heat release.

14 ● Name five types of gas-fired space heaters.

a. Parlor furnaces or circulators.

b. Radiant heaters.

c. Gas-fired steam or hot water radiators.

d. Warm air radiators.

e. Garage heaters.

15 ● How are gas heating units rated?

Gas-fired units are rated on the basis of output in Btu per hour.

16 ● What safety consideration is noted in establishing the ratings of gas-fired units?

The rating is limited by the amount of gas that can be burned without the liberation of harmful amounts of carbon monoxide.

Chapter 12

HEAT AND FUEL UTILIZATION

Total Heat Loss Requirements, Utilization Factors, Degree-Day Methods, Base Temperature Determinations, Steam Consumption of Buildings, Fuel Consumption, Maximum Demands, Load Factors

THE hourly heat loss (H) is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i) of the rooms or spaces to be heated. The total equivalent heating surface required is equal to $\frac{H}{240}$ sq ft.

In estimating the fuel consumption of a building of more than one room divided by walls or partitions, it is not correct to use the calculated heat loss of the building without making the proper allowances for the fact that the heating load at any time does not involve the sum of the infiltration losses of all the heated spaces of the building but only part of the infiltration losses. This is explained in Chapter 6.

It is sufficiently accurate in most cases to consider only half of the total infiltration losses of a building having interior walls and partitions. The value of H in Equation 1 would, under these conditions, be equal to $H_t + \frac{H_i}{2}$. In some cases, where the building has no interior walls or partitions, the infiltration losses are calculated by using only half of the total crack. In this case the entire infiltration loss should be considered.

The heat required to warm the cold building and contents is a factor to be considered. Under certain conditions the cooling of the structure and contents will, to some extent, compensate for the heat required to rewarm the building. For example, if the building is under thermostatic control and the day and night temperatures are, 70 F and 50 F respectively, there will be a period during which no heat will be added while the building is cooling to 50 F, and the saving resulting therefrom will correspond to the additional heat required to bring the building and contents back to the daytime temperature.

ESTIMATING FUEL CONSUMPTION

There are two methods in use for estimating heat or fuel consumption. One method is theoretical, based on a calculated heat loss and assuming absolute constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, sun effect, poor heating systems, etc.

The second method is based on steam consumption data which have been taken from a group of buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the first mentioned method, it is of more value for practical use. Calculations of heat consumption made by the second method will invariably be higher than calculations made by the first method.

Theoretical Estimation Method

To predict the amount of fuel likely to be consumed in heating a building during a normal heating season, it is necessary to know the total heat requirements of the building and the utilization factor of the fuel. The accuracy of the estimate will depend on the ability to select these values and on the care taken in making allowances for other variable factors.

Heat requirements are given by the following general formula:

$$M = \frac{H\,(t - t_a)\,N}{t_d - t_o} \tag{1}$$

Steam requirements are determined by dividing the above by 1000, thus:

$$S = \frac{H\,(t - t_a)\,N}{(t_d - t_o)\,1000} \tag{2}$$

Fuel requirements may be determined by the following formula:

$$F = \frac{M}{C \times E} \tag{3}$$

where

t = inside temperature, degrees Fahrenheit.
t_d = inside design temperature, degrees Fahrenheit.
t_a = average outside temperature, degrees Fahrenheit (Table 2, Chapter 7).
t_o = outside design temperature, degrees Fahrenheit.
H = calculated heat loss of building based on outside temperature (t_o), Btu per hour.
N = number of heating hours per season; 5088 from October 1 to May 1[1].
M = heat loss, Btu per season.
S = steam required to supply M Btu of heat loss.
F = quantity of fuel required per heating season.
C = calorific value of one unit of fuel, the unit being the same as that on which F is based.
E = efficiency of utilization of the fuel, per cent.

Example 1. A small factory building located in Philadelphia is to be heated to 60 F between the hours of 7 A.M. and 7 P.M., and to 50 F during the remaining hours. The calculated hourly heat loss based on a design temperature of −6 F is 500,000 Btu. If coal having a calorific value of 12,500 Btu is fired and the overall heating efficiency is assumed to be 60 per cent, how many pounds of steam would be required for a normal heating season?

[1]This is the period for which t_a (Table 2, Chapter 7) is calculated. If the heating season is different than this period, the corrected values may be substituted for N and t_a.

Solution. Since there are no partitions in the building, the entire heat loss is considered. From Table 2, Chapter 7, the average outside temperature (t_a) during the heating season is 42.7 F; N for the period for which t_a is taken (October 1 to May 1) is 5088; H = 500,000; t_o = −6 F; t = 50 F and 60 F; t_d = 60 F; E = 60 per cent average for heating season; C = 12,500.

The average daily temperature for the 24 hours is:

$$\frac{50 \times 12 + 60 \times 12}{24} = 55 \text{ F}$$

Substituting in Equation 1:

$$M = \frac{500{,}000 \times (55 - 42.7) \times 5088}{60 - (-6)} = 474{,}100{,}000$$

S = 474,100 lb of steam.

$$F = \frac{474{,}100{,}000}{0.60 \times 12{,}500} = 63{,}200 \text{ lb of coal} = 31.5 \text{ tons.}$$

Practical or Degree-Day Method

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association*[2] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. The degree day is defined in Chapter 45. Degree-days for various cities in the United States and Canada are given in Table 1.

Establishing the Base Inside Temperature. Recently the *National District Heating Association* has studied the metered steam consumption of 163 buildings[3] in 22 different cities and has published data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct.

The steam consumption of each building by months was divided by the number of days in each month, thus giving the average daily steam consumption by months. The average steam consumption was then plotted against the average monthly temperature, as shown in Fig. 1, and the temperature at which a line drawn through the points crossed the base line indicated the temperature corresponding to zero steam consumption, or the base temperature. The composite results from 163 buildings calculated in this manner are shown in Table 2.

The resultant average of 66.0 F is close to the *A.G.A.* figure of 65 F. It will be noted that the base temperature calculated for hotels, apartments and residences is consistently higher than those for such buildings as garages, auto sales buildings, and manufacturing buildings. This, of course, would be expected in view of the higher inside temperatures carried in the former group; in fact, an even greater difference would be

[2]See *Industrial Gas Series, House Heating (third edition)* published by the *American Gas Association.*
[3]These buildings are all served with steam from a district heating company.

TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

STATE	CITY	JAN.	FEB.	MAR.	APR.	MAY	SEPT.	OCT.	NOV.	DEC.	TOTAL
Ala.	Birmingham	589	577	260	69				318	595	2408
	Mobile	428	311	152					186	394	1471
Ariz.	Flagstaff	1153	969	896	654	636[b]	292[d]	577	840	1128	7145
	Tucson	459	325	257	87				252	465	1845
Ark.	Hot Springs										2665
	Little Rock	719	582	353	78			47	381	651	2811
Calif.	Los Angeles	326	266	239	159	90			123	301	1504
	San Francisco	465	356	354	294	458[b]	502[c]	146	261	428	3264
Col.	Colorado Springs	1085	993	884	612	459[b]	162	502	789	1067	6553
	Denver	1079	918	799	534	267	72	428	759	1017	5873
Conn.	New Haven	1110	1011	899	543	223	39	360	693	1017	5895
D. C.	Washington										4626
Fla.	Jacksonville	285	207	56					75	267	890
Ga.	Atlanta	682	558	388	132			96	396	639	2891
	Savannah	409	316	167					201	397	1490
Idaho	Boise	1098	848	651	435	235	108	434	738	1011	4558
	Lewiston										4924
Ill.	Chicago	1262	1095	909	549	248	30	353	756	1113	6315
	Springfield	1180	1008	760	365	56		282	681	1038	5370
Ind.	Evansville	949	854	640	276			155	528	862	4164
	Indianapolis	1128	969	756	384	58		298	687	1017	5297
Iowa	Des Moines	1392	1173	890	429	118		357	798	1216	6373
	Sioux City	1434	1386	967	489	164	33	415	870	1265	7023
Kans.	Dodge City	1116	890	688	342	46		276	672	1004	5034
	Topeka	1221	980	741	339			270	699	1051	5301
Ky.	Lexington	974	867	648	342	25		245	612	903	4616
	Louisville	939	801	589	264			186	552	849	4180
La.	New Orleans	332	230	58					102	301	1023
Me.	Eastport	1380	1232	1110	786	843[b]	566[c]	543	843	1228	8531
	Portland	1321	1168	1017	642	368	120	443	780	1153	7012
Md.	Baltimore	955	843	700	348	22		223	567	875	4533
Mass.	Boston	1150	1042	908	570	245	48	363	693	1026	6045
	Springfield										6464
Mich.	Detroit	1253	1134	976	573	226	42	400	777	1113	6494
	Marquette	1501	1360	1249	804	682[b]	268[d]	567	960	1301	8692
Minn.	Duluth	1727	1473	1277	810	722[b]	298[d]	620	1062	1491	9480
	Minneapolis	1609	1400	1095	570	235	93	481	963	1405	7851
Miss.	Vicksburg	520	384	195					252	471	1822
Mo.	Kansas City	1201	987	750	321	15		285	605	1038	5202
	St. Louis	1060	854	657	276			205	597	936	4585
Mont.	Billings	1316	1120	955	534	376[b]	189	524	909	1192	7115
	Havre	1624	1450	1168	630	513	270	620	1041	1383	8699
Nebr.	Lincoln										6231
	Omaha	1355	1125	868	414	84		328	780	1174	6128
Nev.	Reno	1041	823	753	534	456[b]	144	452	714	974	5891
N. H.	Concord	1349	1240	1011	669	351[b]	168	484	846	1234	6852
N. J.	Atlantic City	992	903	806	519	220		254	588	893	5175
	Trenton	1014	942	735	402	81		242	588	930	4934
N. M.	Santa Fe	1110	902	775	543	301[b]	120	459	780	1073	6063
N. Y.	Albany	1286	1142	980	549	493	72	446	774	1147	6889
	Buffalo	1240	1156	1032	675	347	75	418	774	1104	6821
	New York	1061	960	837	486	155		276	618	955	5348
	Utica	1242	1181	991	587	244	187	426	787	1140	6785
N. C.	Raleigh	722	630	446	183			130	429	694	3234
	Wilmington	555	468	322	108			19	303	527	2302
N. Dak.	Bismarck										8498

[a]Heating and Ventilating Degree-Day Handbook.
[b]Including June.
[c]Including July and August.
[d]Including August.

TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a] (Continued)

STATE	CITY	JAN.	FEB.	MAR.	APR.	MAY	SEPT.	OCT.	NOV.	DEC.	TOTAL
Ohio	Cincinnati	1076	910	747	378	73	......	290	675	980	5129
	Cleveland	1180	1075	950	564	220	27	366	732	1060	6154
	Columbus	1113	980	703	420	87	......	313	690	1017	5323
Okla.	Oklahoma City	865	742	465	162	......	......	105	459	815	3613
Oreg.	Portland	806	644	558	402	335[b]	105	332	558	728	4468
	Salem	......	......	......	......	......	......	......	......	......	4629
Pa.	Philadelphia	1001	895	756	402	68	......	242	588	903	4855
	Pittsburgh	1054	944	787	423	78	......	313	669	967	5235
R. I.	Providence	1116	1069	890	558	251	63	348	693	1026	6014
S. C.	Charleston	487	372	242	36	......	......	......	207	425	1769
	Spartanburg	725	688	431	147	......	......	121	429	716	3257
S. Dak.	Sioux Falls	......	......	......	......	......	......	......	......	......	7683
Tenn.	Memphis	744	599	384	96	......	......	62	402	663	2950
	Nashville	812	747	476	180	......	......	136	483	744	3578
Texas	Austin	......	......	......	......	......	......	......	......	......	1578
	Dallas	......	......	......	......	......	......	......	......	......	2455
	Houston	366	277	65	......	......	......	......	114	335	1157
	San Antonio	381	274	74	......	......	......	......	126	347	1202
Utah	Logan	1260	1072	893	525	376	114	468	819	1218	6735
	Salt Lake City	1110	885	722	453	234	18	388	723	1020	5553
Vt.	Burlington	1535	1294	1089	654	276[b]	144	481	861	1286	7620
Va.	Fredericksburg	887	820	583	303	......	......	223	549	878	4243
	Norfolk	738	650	520	246	......	......	99	411	685	3349
	Richmond	825	702	552	240	......	......	158	483	765	3725
Wash.	Seattle	775	653	623	465	487[b]	276[c]	403	570	716	4968
	Spokane	1171	952	778	504	366	192	514	819	1057	6353
W. Va.	Morgantown	1026	944	713	414	......	......	294	648	977	5016
	Parkersburg	......	......	......	......	......	......	......	......	......	4884
Wis.	Fond du Lac	1507	1321	1046	603	276	117	493	921	1328	7612
	Green Bay	1538	1358	1125	600	322	132	505	921	1322	7823
	LaCrosse	1535	1265	1032	528	183	96	462	909	1280	7290
	Milwaukee	1383	1328	1023	648	389[b]	84	449	846	1222	7372
Wyo.	Cheyenne	1215	1075	995	720	569	240	605	900	1143	7462

PROVINCE	CITY	JAN.	FEB.	MAR.	APR.	MAY	SEPT.	OCT.	NOV.	DEC.	TOTAL
B. C.	Victoria	......	......	......	......	......	......	......	......	......	5777
	Vancouver	......	......	......	......	......	......	......	......	......	5976
	Kamloops	......	......	......	......	......	......	......	......	......	6724
Alb.	Medicine Hat	......	......	......	......	......	......	......	......	......	8152
Sask.	Qu'Appelle	......	......	......	......	......	......	......	......	......	11,261
Man.	Winnipeg	......	......	......	......	......	......	......	......	......	11,166
Ont.	Port Arthur	......	......	......	......	......	......	......	......	......	10,803
	Toronto	......	......	......	......	......	......	......	......	......	7732
Que.	Montreal	1615	1409	1219	720	309	190	372	961	1422	8417
	Quebec	......	......	......	......	......	......	......	......	......	8628
N. B.	Fredericton	......	......	......	......	......	......	......	......	......	9099
N. S.	Yarmouth	......	......	......	......	......	......	......	......	......	7694
P. E. I.	Charlottetown	......	......	......	......	......	......	......	......	......	8485

[a]Heating and Ventilating Degree-Day Handbook.

[b]Including June.

[c]Including July and August.

TABLE 2. BASE TEMPERATURE FOR THE DEGREE-DAY[a]

Type of Building	No. of Buildings Analyzed	Temperature F Corresponds to Zero Steam Consumption
Office	60	66.2
Office and Bank	4	65.8
Bank	3	66.2
Office and Telephone Exchange	2	65.5
Office and Stores	6	67.4
Stores	11	64.0
Department Stores	12	64.3
Hotels	7	66.5
Apartments	14	68.8
Residences	8	66.9
Clubs	4	65.5
Lodges	5	64.9
Theatres	3	67.6
Churches	2	65.8
Garage	2	64.8
Auto Sales and Service	4	61.2
Newspaper and Printing	3	67.7
Warehouse and Loft	3	67.7
Office and Loft	2	65.2
Manufacturing	8	65.4
Average for 163 Buildings		66.0 F

[a]Report of *Commercial Relations Committee, 1932 Proceedings, National District Heating Association.*

TABLE 3. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS[a]
(Heating Season Only)

Building Classification	No. of Buildings Listed	Steam Consumption Pounds per Degree-Day—65 F Basis[d]		
		Per M Cu Ft of Heated Space	Per M Sq Ft of Radiator[c] Surface	Per M Btu per Hr of Heat Loss[b]
Apartments	16	1.78	97.5	0.359
Hotels	10	1.46	80.6	0.371
Residences	12	1.32	64.2	....
Printing	7	1.25	105.5	....
Clubs and Lodges	10	0.96	77.0	....
Retail Stores	18	0.90	80.6	0.268
Theatres	6	0.90	75.0	0.498
Loft and Mfg.	16	0.89	72.3	0.283
Banks	7	0.88	45.2	....
Auto Sales and Service	8	0.83	62.2	....
Churches	6	0.58	49.4	....
Department Stores	14	0.57	60.7	0.238
Garages (Storage)[e]	6	0.42	72.3	....
Offices (Total)	35	1.09	70.0	0.283
Offices (Heating only)	35	0.975	65.4	0.256

[a]Includes steam for heating domestic water for heating season only.
[b]Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).
[c]Equivalent steam radiator surface.
[d]The figures are a numerical—not a weighted—average for the several buildings in each class.
[e]Based on zero consumption at 55 F.

expected. For an average figure, the *A.G.A.* base of 65 F may therefore be safely used, and if greater refinement is desired, the figure for the type of building under consideration can be taken from Table 2.

Table 3[4] gives the steam consumption per degree-day, expressed in three different ways, for 196 buildings in 14 different classifications. These buildings are divided among 21 different cities in the United States. The steam used for heating the domestic water is included in these figures, but in the case of office buildings, the steam for heating only is also shown. The data are placed on a comparable basis by expressing the steam consumption in terms of pounds per degree-day per thousand square feet of equivalent installed radiator surface, per thousand cubic feet of heated space, and per thousand Btu of calculated heat loss.

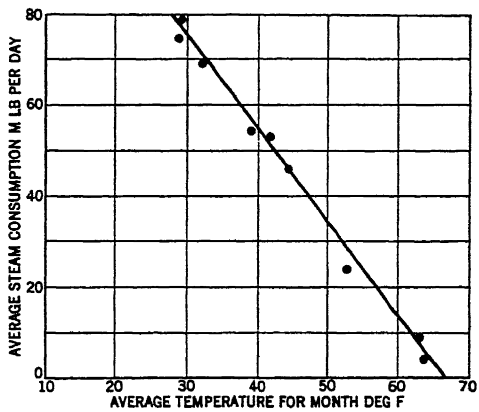

FIG. 1. METHOD OF DETERMINING BASE TEMPERATURE FOR DEGREE-DAY CALCULATIONS[5]

The choice of these units of comparison require some explanation. The use of *heated* space in preference to the gross cubage used by architects is obviously more accurate for this purpose. The architect's cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. The cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents.

The use of radiator surface as the basis of comparison has two objections. One is that the amount of radiator surface in a building is often either excessive or deficient, and figures for steam consumption based on it are therefore likely to be in error. Another reason is that it is difficult to convert fan coil surface into equivalent direct radiator surface with accuracy. On the whole, the use of radiator surface as the basis of comparison is the least satisfactory of the three methods.

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[4]The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 171).

[5]Report of *Commercial Relations Committee, 1932 Proceedings, National District Heating Association.*

It should be noted that the figures in Table 3 are for the heating season only and include steam for heating domestic water.

Example 1 solved by the degree-day method and using values taken from Table 3 would show a higher steam consumption.

Example 2. Factor for steam consumption for a manufacturing building per M Btu per hour heat loss per degree-day = 0.283; total number of degree-days per year (Table 1) = 4855; heat loss = 500,000 Btu per hour.

Solution. 0.283 × 4855 × 500 = 686,982 lb of steam per year. This calculation results in an estimate 45 per cent higher than the previous calculation and one which would be more nearly correct for actual practice.

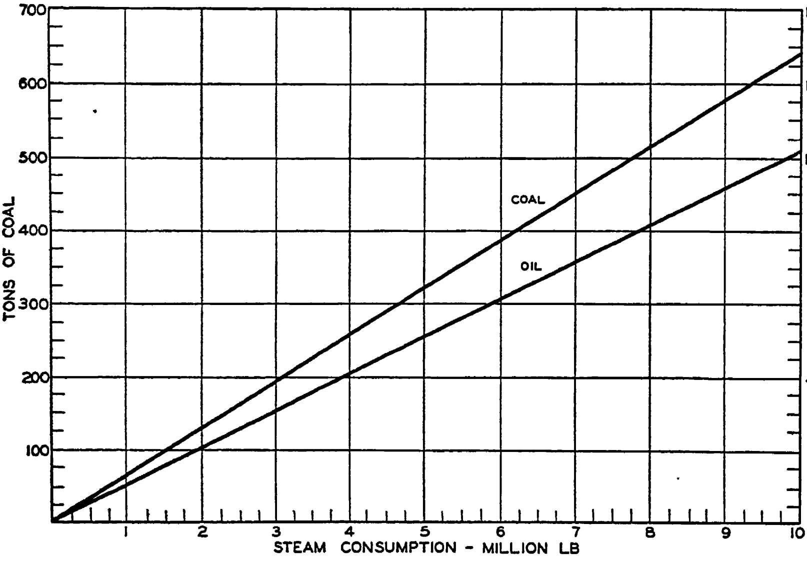

FIG. 2. CURVE FOR ESTIMATING FUEL CONSUMPTION FOR VARIOUS KNOWN STEAM CONSUMPTION[a]

[a]This curve is based on heating efficiencies of 60 to 70 per cent for coal and oil, respectively, a calorific value of coal of 13,000 Btu per pound, a calorific value of oil of 140,000 Btu per gallon.

In case the heat loss figure is not known this method of estimating heat or steam consumption can also be applied if the net heated space figure is available.

CALCULATION OF FUEL CONSUMPTION

After the heat and steam consumption of the building have been calculated, the corresponding fuel requirements may also be estimated by assuming the correct boiler and furnace efficiencies. If the building is to be supplied with steam from a district heating company, the steam consumptions as calculated by the two Methods are generally assumed to be correct. However, if the steam is to be supplied from an individual boiler

unit, the consumption should be assumed as from 10 to 20 per cent greater. One reason for this difference in steam consumption is that district steam is a metered service, and building managers are therefore more conscious of their heating costs, which generally results in better maintained heating systems. Also, the district steam service is usually installed with thermostatic control which reduces overheating to a minimum.

Fig. 2 shows the amount of coal or oil that may be estimated when the steam consumption is known. Assuming the steam consumption that

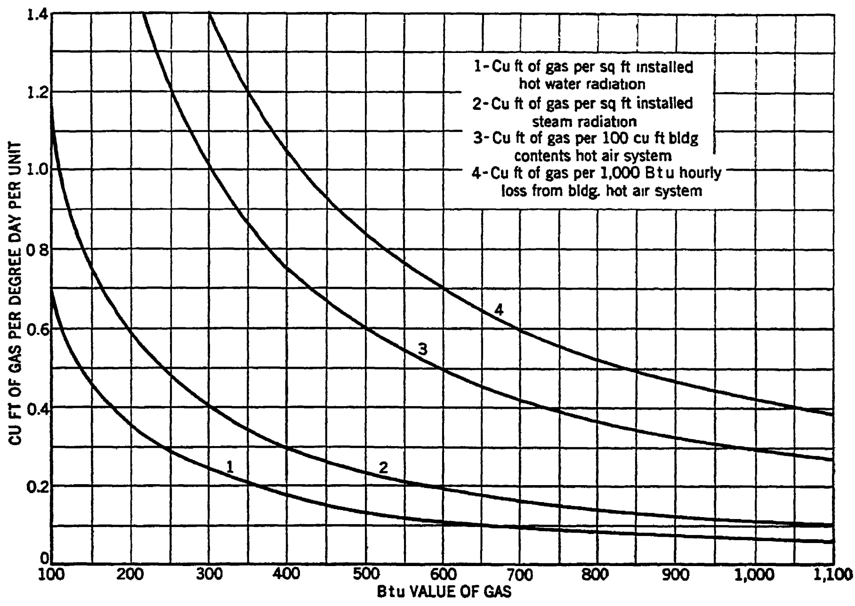

FIG. 3. CHART GIVING GAS REQUIREMENTS PER DEGREE-DAY FOR VARIOUS CALORIFIC VALUES OF GAS AND FOR DIFFERENT HEATING SYSTEMS[a]

[a]This chart is based on an inside temperature of 70 F and an outside temperature of zero. If the *radiation* is installed on the basis of any other temperature difference, multiply the result obtained from this chart by 70, and divide by the actual temperature difference. From *Industrial Gas Series House Heating* (*third edition*) published by the *American Gas Association.*

was calculated in Example 2, 686,982 lb, the corresponding coal consumption, from the curve, is 44 tons and the oil consumption 6000 gal.

Fig. 3 indicates the average gas consumption per degree-day for various heat contents. While the fuel consumption in individual cases may vary somewhat from the curve values, these average values are sufficiently accurate for estimating purposes and give satisfactory results.

The value generally used in the manufactured gas industry for residences is 0.21 cu ft per degree-day per square foot of equivalent steam *radiation* (240 Btu) based on the theoretical requirements. A correction for warmer climates is necessary and it is customary to gradually increase the relative fuel consumption below 3000 degree-days to about 20 per cent more at 1000 degree-days.

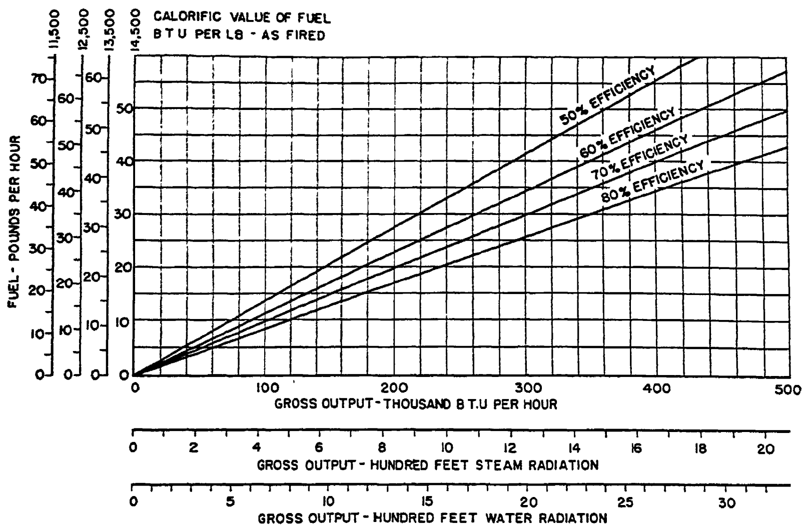

FIG. 4. COAL FUEL BURNING RATE CHART.

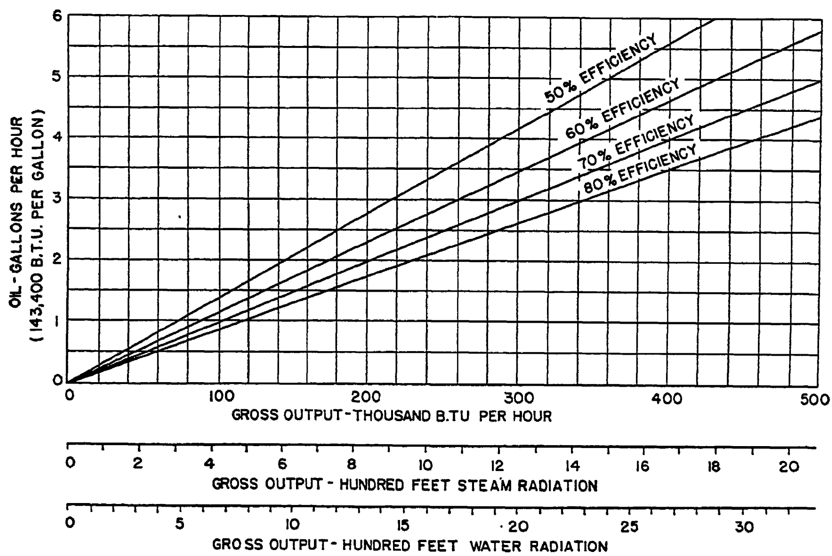

FIG. 5. OIL FUEL BURNING RATE CHART.[a]

[a]This chart is based upon No. 3 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 2 oil (141,000 Btu per gallon) 1.017; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956

For hot water or warm air heat the fuel consumption is about 0.19 cu ft per degree-day per square foot of equivalent steam *radiation*, that is, per 240 Btu per hour. The actual requirements likewise relatively increase with hot water or warm air systems as the number of degree-days decreases below 3000. For larger installations, that is 1000 sq ft of theoretical *radiation* and above, there is an increase in efficiency, and a consequent decrease in the fuel consumption per degree-day per square foot of heating surface.

The approximate quantities of steam required in New York City per square foot of heating surface for various classes of buildings are given in Chapter 42.

The preceding discussion on fuel consumption has dealt with the heating requirements of the building irrespective of any air that may be

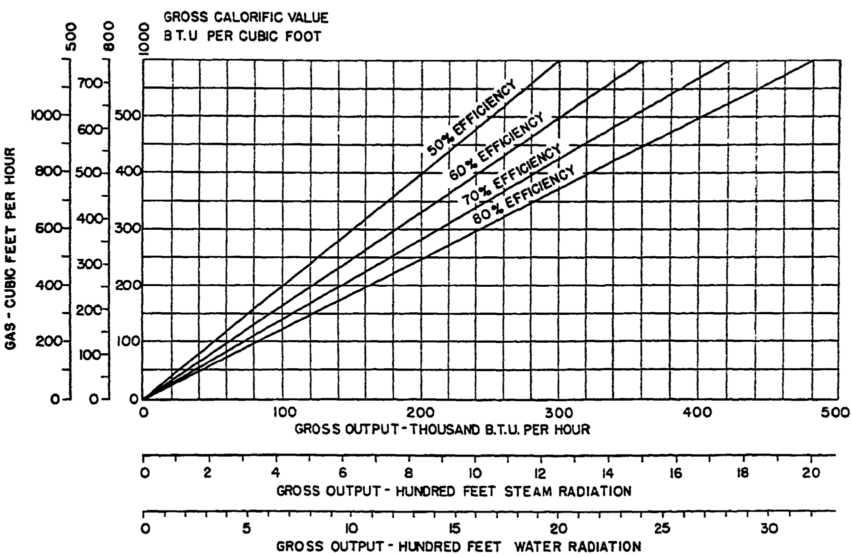

FIG. 6. GAS FUEL BURNING RATE CHART.

introduced for ventilation purposes other than the normal infiltration of outside air. The heat required for warming air brought into the building for ventilation may be estimated from data given in Chapters 3 and 21.

Rate of Fuel Burning

If the estimated maximum load or gross output is determined as outlined in Chapter 13, the fuel burning rates for coal, oil, or gas may be determined from the charts illustrated in Figs. 4, 5 and 6. For a given efficiency, the rate of fuel burning is directly proportional to the gross output, and therefore, these charts can be extended by moving the decimal point the same number of digits in both vertical and horizontal scale. All charts are based upon values of one square foot steam radiation equivalent to 240 Btu per hour and one square foot hot water radiation equivalent to 150 Btu per hour. In using these charts, consideration

must be given to the overall efficiency of the boiler or firing device. This factor is largely dependent upon good judgment and is a measure of the degree of efficiency to be expected from a particular installation.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross or maximum design load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

In the case of warm air heating installations the heat loss of the building expressed in Btu per hour can be determined by use of the Standard Codes[6] of the *National Warm Air Heating and Air Conditioning Association*. If the design heat loss of the building is divided by the proper factor, the required gross output may be obtained from the proper charts shown in Figs. 4, 5 and 6.

Example 3. The estimated net load (including domestic hot water supply) as calculated for a residence is 1500 sq ft of hot water radiation. Determine the firing rates for various fuels assuming an overall efficiency of 70 per cent; using coal with a calorific value of 12,500 Btu per pound; No. 3 fuel oil and natural gas having a gross heating value of 1000 Btu per cubic foot.

Solution. Referring to Fig. 3, Chapter 13, a piping and pick-up factor for a net load of 1500 sq ft is found to be 43 per cent or the gross output is equivalent to 1500 × 1.43 = 2145 sq ft hot water radiation.

Using the chart in Figs. 4, 5 and 6 project vertically from the gross output value on the proper horizontal scale to the intersection of the 70 per cent efficiency line. From the intersection of this line proceed horizontally to the proper vertical scale where a direct value of the required fuel burning rate is given. These values are rates of burning while firing device is in operation and are not indicative of hourly fuel consumption.

By use of the respective charts the firing rates for the various fuels will be found to be: coal 36.8 lb per hour, oil 3.2 gal per hour, and gas 460 cu ft per hour.

TABLE 4. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS[a]

BUILDING CLASSIFICATION	LOAD FACTOR	LB OF DEMAND PER HR PER SQ FT OF EQUIVALENT INSTALLED RADIATOR SURFACE
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theatres	0.126	0.151

[a]Loc. Cit. Note 5.

[6]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences (9th edition) and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems, may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 50 W. Broad St., Columbus, Ohio.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 4.

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 4, the theatres, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

PROBLEMS IN PRACTICE

1 ● What will be the cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The heating efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 p.m. to 7 a.m. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

The average hourly temperature is

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F}.$$

The maximum hourly heat loss will be

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu}.$$

$$M = \frac{79{,}000\ (66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

$$2204.6 \times 0.07 = \$154.34 = \text{cost per year of heating the building.}$$

2 ● What factors should be taken into consideration when determining the efficiency at which a fuel will be burned?

Manufacturers' catalogs usually give equipment efficiencies obtained under test conditions. These values do not allow for poor attendance, defects in installation, or poor draft. Such efficiencies do not consider heat radiated from the outside of the equipment, but in many cases this heat is utilized.

3 ● If 20 tons of coal having a calorific value of 13,000 Btu per pound are burned in a warm air furnace and produce 286,000,000 Btu at the bonnet, what is the efficiency of the furnace?

$$\frac{\text{Number of Btu at bonnet}}{\text{Number of tons} \times \text{calorific value} \times \text{number of pounds in one ton}} = \text{efficiency.}$$

$$\frac{286{,}000{,}000 \times 100}{20 \times 13{,}000 \times 2000} = 55 \text{ per cent.}$$

4 ● Make a rough approximation of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperatures of 0 F and 70 F. Chicago has 800-Btu mixed gas, and 6315 degree-days.

Using Fig. 3, the fuel consumption for a design temperature of 0 F with 800-Btu gas is found to be 0.08 cu ft of gas per degree-day per square foot of hot water radiation.

$$0.08 \times 6315 \times 1000 = 505{,}200 \text{ cu ft.}$$

5 ● A certain building has a maximum heat loss of 250,000 Btu per hour in −15 F weather. How many tons of fuel will be required to maintain a temperature of 70 F during a 260-day heating season in which the average temperature is 39 F? The heating value of the fuel is 13,200 Btu per pound and the efficiency of combustion is 60 per cent.

$$\frac{250{,}000\ (70 - 39)\ 260 \times 24}{(70 + 15)\ 13{,}200 \times 0.60 \times 2000} = 35.9 \text{ tons.}$$

6 ● Which item may be determined more closely, the heating value of a fuel or the efficiency of its combustion?

The heating values of oil, gas, and solid fuels are closely determinable, whereas the efficiency of burning depends on the particular equipment chosen and the skill used in handling it.

7 ● In an office building, the thermostats are set to maintain 70 F from 7 a.m. to 5 p.m. and 50 F during the rest of the time. When the outside temperature is 30 F, how much saving might be expected because the temperatures are lowered? Under the above conditions the building becomes 50 F by 11 p.m. and warms up to 70 F by 8 a.m.

A temperature of 70 F is maintained during 9 hours, and one of 50 F during 8 hours; the temperature would average about 60 F during the 7 hours required for cooling down and warming up. The average is 60.4 for the 24 hours. (The average temperature calculated would have been 58.3 F, had the warming and cooling periods been neglected.)

The saving is $\left(\frac{70 - 60.4}{70 - 30}\right) \times 100 = \frac{9.6}{40} \times 100 = 24$ per cent.

8 ● How does the heat capacity of a structure influence the saving made by carrying lower temperatures during the night?

The heat storage capacity of the walls prevents rapid dropingof temperatures at night-time and delays the warming up process in the morning. In an extreme case, the building would not reach the lowered temperature by the time the higher temperature is called for in the morning. But under any conditions, the saving made by lowering the temperature can be correctly estimated by using the average temperature observed over the 24-hour period as a factor, as in Question 7.

9 ● What are some of the miscellaneous factors that may cause actual fuel consumption to vary from the theoretical fuel requirements as calculated by the use of heat losses, temperature difference, and fuel burning efficiency?

The opening of windows; abnormally high or low inside temperatures; other sources of heat, such as machinery or lights; sun effect; high occupancy; and unusual winds.

Chapter 13

HEATING BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section*, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers may be of round pattern with circular grate and horizontal pancake sections joined by push nipples and tie rods, or of rectangular pattern with vertical sections. The latter type may be either of outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers, or assembled with push nipples and tie rods, in which case the water and steam connections are internal.

Cast-iron boilers usually are shipped knocked down to facilitate handling at the place of installation where assembly is made. One of the chief advantages of cast-iron boilers is that the separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature is of importance in making repairs to or replacing a damaged or worn-out boiler and should be given consideration in the original selection. Sufficient space should be provided in the boiler room for assembling the boiler and for disassembling it conveniently if repairs are needed. With the outside header type of boiler a damaged section in the middle of the boiler can be removed without disturbing the other sections so side clearance should be provided.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of steam radiation. For larger loads, cast-iron boilers must be installed in multiple, or a steel boiler must be used. In most cases cast-iron boilers are limited to working pressures of 15 lb for steam and 30 lb for water. Special types are built for hot water supply which will withstand higher local water pressures.

STEEL BOILERS

Two general classifications may be applied to steel boilers: *first*, with regard to the relative position of water and hot gases, distinguished as fire tube or water tube; *second*, with regard to arrangement of furnace and

flues, as (1) horizontal return tubular (HRT) boilers, (2) portable (self-contained) firebox boilers with either water or fire tubes, and (3) water tube boilers of the power type.

Fire tube boilers are constructed so that the water available to produce steam is contained in comparatively large bodies distributed outside of the boiler tubes, the hot gases passing within the tubes. In *water tube* boilers, the water is circulated within the boiler tubes, heat being applied externally to them.

The *HRT boiler* is the oldest type and consists of a horizontal cylindrical shell with fire tubes, enclosed in brickwork to form the furnace and combustion chamber. All heating surfaces and the interior of the boiler are accessible for both cleaning and inspection. Horizontal return tubular boilers, especially the larger sizes, should be suspended from structural columns and beams independent of the brick setting. Small HRT boilers sometimes are supported by brackets resting on the brick setting.

TABLE 1. PRACTICAL COMBUSTION RATES FOR SMALL COAL-FIRED HEATING BOILERS OPERATING ON NATURAL DRAFT OF FROM 1/8 IN. TO 1/2 IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

Portable firebox boilers are the more generally used type of steel heating boilers, their outstanding characteristic being the water-jacketed firebox which eliminates virtually all brickwork. They are shipped in one piece from the factory and come to the job ready for immediate hook-up to piping. They may be of welded or riveted construction and have either water or fire tubes. Manufacturers' catalogs usually list heating surface as well as grate area. The elimination of brickwork also makes this type the most compact of steel boilers as well as the lowest in first cost.

Water tube boilers. For large heating loads water tube boilers are quite frequently used. They usually require more head room than other types of boilers but require considerably less floor space and make possible a

much higher rate of evaporation per square foot of heating surface, with proper setting, baffling and draft. Water tube boilers used for heating purposes are either completely supported, insulated and encased in steel, or else brick set, supported on structural steel columns and have the brick setting encased in an insulated steel housing to prevent air infiltration and to minimize heat losses. For large heating loads at a high rate of evaporation, such boilers should be operated at pressures above 15 lb per square inch with a pressure-reducing valve on the connection to the heating main.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite and coke. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to meet the general requirements of oil burners or are specially adapted to one particular burner have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal fired boiler converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low. See Chapter 11.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or

where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 11.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. Investigations[1] by the U. S. Bureau of Mines show that:

1. A boiler in which the heating surface is arranged to give long gas passages of small cross-section will be more efficient than a boiler in which the gas passages are short and of larger cross-section.
2. The efficiency of a water tube boiler increases as the free area between individual tubes decreases and as the length of the gas pass increases.
3. By inserting baffles so that the heating surface is arranged in series with respect to the gas flow, the boiler efficiency will be increased.

The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about 3300 Btu per sq ft per hour for hand-fired boilers and 4000 Btu per sq ft

[1]See *U. S. Bureau of Mines Bulletin* No. 18, The Transmission of Heat into Steam Boilers.

per hour for mechanically fired boilers when operating at *design load.* When operating at *maximum load*[2] these values will run between 5000 and 6000 Btu per sq ft per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted three solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[3], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[3] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[4]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[5] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

Steel Heating Boilers Ratings

The *Steel Heating Boiler Institute* has adopted a method for the rating of low pressure boilers based on their physical characteristics and expressed in square feet of steam or water radiation or in Btu per hour as given in Table 2. The following requirements are included in this Code:

1. One square foot of steam radiation is to be considered equal to the emission of 240 Btu per hour and one square foot of water radiation is to be considered equal to emission of 150 Btu per hour.
2. The rating of a boiler expressed in square feet of steam radiation in which solid fuel hand fired is used is based on the amount equal to 14 times the heating surface of the boiler in square feet.
3. The rating of a boiler expressed in square feet of steam radiation in which solid fuel mechanically fired, or in which oil or gas is burned, is based on the amount equal to 17 times the heating surface of the boiler in square feet.
4. Heating surface is to be expressed in square feet and include those surfaces in the boiler which are exposed to the products of combustion on one side and water on the other. In measuring surfaces, the outer tube areas are to be considered. When a boiler has the water leg height increased the heating surface noted in the published ratings are not to be increased.
5. A grate area is to be considered as an area of the grate surface expressed in square feet and measured in the plane of the top surface of the grate. For double grate boilers the grate surface is to be considered as the area of the upper grate plus one-quarter of the area of the lower grate.
6. The grate area of a boiler for rating as determined in No. 2 is to be not less than that determined by the following formulae:

For boilers with ratings 1800 sq ft to 4000 sq ft of steam radiation:

$$\text{Grate Area} = \sqrt{\frac{\text{Catalogue Rating (in square feet steam radiation)} - 200}{25.5}} \qquad (1)$$

[2]For definitions of design load and maximum load see pages 251 and 252.
[3]See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 12. Also Chapter 45.
[4]See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35. Also Chapter 45.
[5]See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 23. Also Chapter 45.

TABLE 2. STANDARD STEEL HEATING BOILER RATINGS[a]

Hand Fired Capacity Rating					Mechanically Fired Capacity Rating			
Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hr	Heating Surface Sq Ft	Grate Area Sq Ft	Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hr	Furnace Volume Oil, Gas or Bituminous Coal Cu Ft
1,800	2,880	432,000	129	7.9	2,190	3,500	525,600	15.7
2,200	3,520	528,000	158	8.9	2,680	4,280	643,200	19.2
2,600	4,160	624,000	186	9.7	3,160	5,050	758,400	22.6
3,000	4,800	720,000	215	10.5	3,650	5,840	876,000	26.1
3,500	5,600	840,000	250	11.4	4,250	6,800	1,020,000	30.4
4,000	6,400	960,000	286	12.2	4,860	7,770	1,166,400	34.8
4,500	7,200	1,080,000	322	13.4	5,470	8,750	1,312,800	39.1
5,000	8,000	1,200,000	358	14.5	6,080	9,720	1,459,200	43.5
6,000	9,600	1,440,000	429	16.4	7,290	11,660	1,749,600	52.1
7,000	11,200	1,680,000	500	18.1	8,500	13,600	2,040,000	60.8
8,500	13,600	2,040,000	608	20.5	10,330	16,520	2,479,200	73.8
10,000	16,000	2,400,000	715	22.5	12,150	19,440	2,916,000	86.8
12,500	20,000	3,000,000	893	25.6	15,180	24,280	3,643,200	108.5
15,000	24,000	3,600,000	1,072	28.4	18,220	29,150	4,372,800	130.2
17,500	28,000	4,200,000	1,250	30.9	21,250	34,000	5,100,000	151.8
20,000	32,000	4,800,000	1,429	33.2	24,290	38,860	5,829,600	173.5
25,000	40,000	6,000,000	1,786	37.4	30,360	48,570	7,286,400	216.9
30,000	48,000	7,200,000	2,143	41.2	36,430	58,280	8,743,200	260.3
35,000	56,000	8,400,000	2,500	44.7	42,500	68,000	10,200,000	303.6

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendation R 157-35.*

For boilers with ratings 4000 sq ft of steam radiation and larger:

$$\text{Grate Area} = \sqrt{\frac{\text{Catalogue Rating (in square feet steam radiation)} - 1500}{16.8}} \quad (2)$$

7. The volume for furnaces in which solid fuel is burned is to be considered as the cubical content of the space between the bottom of the fuel bed and the first plane of entry into or between the tubes. Volume of furnaces in which pulverized liquid fuel or gaseous fuel is burned are to be considered as the cubical content of the space between the hearth and the first plane of entry into or between the tubes. No minimum furnace volume is to be specified for mechanical fired boilers burning anthracite.

8. The furnace volume for a boiler, with a rating as determined in No. 3 in which oil, gas or bituminous coal stoker fired is burned is not to be less than one cubic foot for every 140 sq ft of steam rating.

9. The average height of furnace for the rating determined in No. 3 in which bituminous coal, stoker fired is burned is not to be less than that determined graphically in Fig. 1 or mathematically by the following formula:

$$H = \sqrt{\frac{R}{22.5}} + \sqrt{\frac{R}{A}} \quad (3)$$

where

H = average furnace height, inches as determined by the following formula;

$$H = \frac{12F}{A} = \frac{12F}{WL}$$

R = stoker fired boiler rating, square foot steam radiation.
A = plan area of firebox, square feet measured at the bottom of the fuel bed.
F = furnace volume, cubic feet.

W = average width of furnace, measured at the bottom of the fuel bed, feet.
L = length of furnace, feet. If the furnace is longer than the fuel bed or contains a bridge wall, the total length of the furnace may be used except that this length is not to exceed $2\frac{1}{2}$ W.

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for

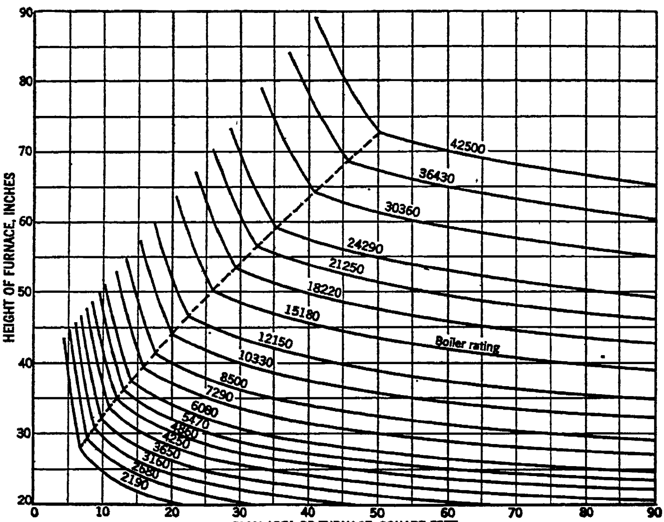

FIG. 1. FURNACE HEIGHTS FOR STOKER FIRED BOILERS AND BITUMINOUS COAL RATED IN SQUARE FEET STEAM RADIATION

Rating Steam Heating, Solid Fuel, Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of *boiler horsepower* instead of in Btu per hour or square feet of equivalent radiation.

Boiler Horsepower: The evaporation of 34.5 lb of water per hour *from and at 212 F* which is equivalent to a heat output of $970.2 \times 34.5 = 33{,}471.9$ Btu per hour.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at this same temperature and at atmospheric pressure.

It is usually considered that 10 sq ft of boiler heating surface will produce a rated boiler horsepower. A rated boiler horsepower in turn can carry a design load of from 100 to 140 sq ft of equivalent radiation. It is apparent, therefore, that 1 sq ft of boiler heating surface can carry a design load of from 10 to 14 sq ft of equivalent radiation, or somewhat more if the boiler is forced above rating. The application of these values is discussed under the heading Selection of Boilers.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel to the calorific value of 1 lb of fuel.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[6].

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated design load is the sum of the following three items[7]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.

[6]Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 517), A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 317).

[7]A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932.)

The estimated maximum load is given by[8]:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 3.

TABLE 3. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3,[d]		PERCENTAGE CAPACITY TO ADD FOR WARMING UP
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggert (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired, solid fuel boilers. A factor of 20 per cent over design load is adequate when automatically-fired fuels are used (see Fig. 3).

[d]240 Btu per square foot.

Other things to be considered are:

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.
6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.
7. Combustion space in the furnace.
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown and headroom in the boiler room.

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 5, 6 and 7, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 14. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 14.)

Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration in the various spaces of the building to be heated, which reserve, however,

[8]Loc. Cit. Note 7.

is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 6.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The

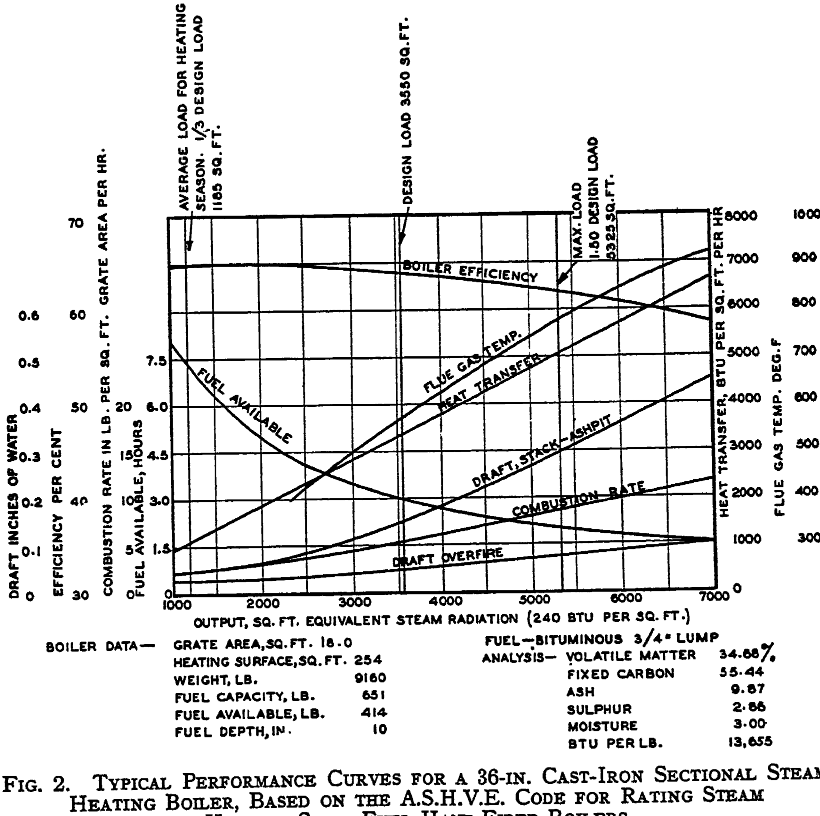

FIG. 2. TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 43.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of 25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply

and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 39. A chart is shown in Fig. 3 indicating percentage allowances for piping and warming-up which are applicable to automatically-fired heating plants using steam radiation. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

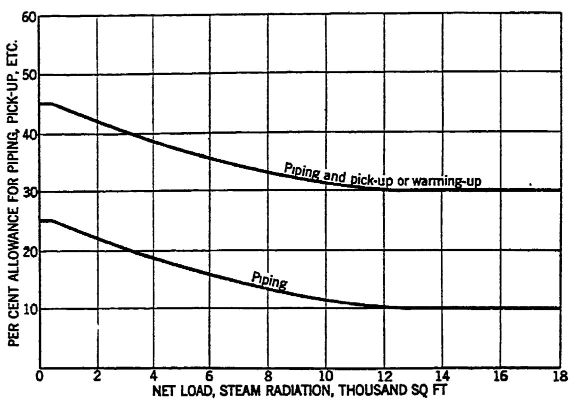

FIG. 3. PERCENTAGE ALLOWANCE FOR PIPING AND WARMING-UP

Warming-Up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the allowance to be made should be selected from Table 3 and should be applied to the estimated design load as determined by Items 1, 2 and 3. While in every case the estimated maximum load will exceed the design load if adequate heating response is to be achieved, there is however, no object in over-estimating the allowances, as the only effect would be to reduce the time of warming-up by a few minutes. Otherwise, it might result in firing the boiler unduly and increasing the cost of operation.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 2. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{4}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 251).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 4. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Steel Heating Boilers

Boiler ratings previously described under the *Steel Heating Boiler Institute's* Boiler Rating Code are intended to correspond with the estimated design load based on the sum of items 1, 2 and 3 outlined on pages 252 and 253. Insulated residence type boilers for oil or gas may carry a net load expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler manufacturer guarantees the boiler to be capable of operating at a maximum output of not less than 150 per cent of net load rating with overall efficiency of not less than 75 per cent with at least two different makes of each type of standard commercial burner recommended by the boiler manufacturer. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the net load.

When the estimated heat emission of the piping (connecting radiation, and other apparatus to the boiler) is not known the net load to be considered for the boiler may be determined from Table 4.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with the percentage allowances given in Fig. 3. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum

TABLE 4. BOILER RATINGS BASED ON NET LOAD[a]

HAND FIRED RATINGS		MECHANICALLY FIRED RATINGS	
Steam Radiation Sq Ft	Net Load[b] Steam Radiation Sq Ft	Steam Radiation Sq Ft	Net Load[b] Steam Radiation Sq Ft
1,800	1,389	2,190	1,695
2,200	1,702	2,680	2,089
2,600	2,020	3,160	2,461
3,000	2,335	3,650	2,853
3,500	2,732	4,250	3,335
4,000	3,135	4,860	3,830
4,500	3,540	5,470	4,330
5,000	3,945	6,080	4,834
6,000	4,770	7,290	5,850
7,000	5,608	8,500	6,885
8,500	6,885	10,330	8,490
10,000	8,197	12,150	10,125
12,500	10,417	15,180	12,650
15,000	12,500	18,220	15,183
17,500	14,584	21,250	17,708
20,000	16,667	24,290	20,242
25,000	20,834	30,360	25,300
30,000	25,000	36,430	30,359
35,000	29,167	42,500	35,417

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards*, *United States Department of Commerce Simplified Practice Recommendation R 157-35.*

[b]The net load is made up by the sum of the estimated design load, items 1 and 2 (pages 252 and 253). All net loads are expressed in 70 F. For hand fired boiler ratings less than 1800 sq ft of steam or 2880 sq ft of water and mechanically fired boiler ratings of 2190 sq ft of steam or 3500 sq ft of water, apply the factor 1.3 to the net load to determine the boiler size. For water boilers use the equivalent net load for steam boilers of similar physical size.

size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is under-estimated or more is added in the future.

Conversions

The conversion of a coal or oil boiler to gas burning is simpler than the reverse since little furnace volume need be provided for the proper combustion of gas. When a solid fuel boiler of 500 sq ft (or less) capacity is converted to gas burning, the necessary gas heat units should be approximately double the connected load. The presumption for a conversion job is that the boiler is installed and probably will not be made larger;

therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. Assuming a combustion efficiency of 75 per cent for a conversion installation the boiler output would be $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pickup. In converting large boilers, the determination of the required Btu input should not be done by an arbitrary figure or factor but should be based on a detailed consideration of the requirements and characteristics of the connected load.

An efficient conversion installation depends upon the proper size of flue connection. Often the original smoke breeching between the boiler and chimney is too large for gas firing, and in this case, flue orifices can be used. They are discs provided with an opening of the size for the gas input used in this boiler. The size should be based on 1 sq in. of flue area for each 7500 hourly Btu input.

If dampers are found in the breeching they should be locked in position so that they will not interfere with the normal operation of the gas burners at maximum flow. In the case of large boiler conversions, automatic damper regulators proportion the position of the flue dampers to the amount of gas flowing and may be substituted for existing dampers. Generally in residence conversions automatic dampers are not of the proportioning type but close the flue during the off periods of the gas burners. Automatic shutoff dampers should be located between the backdraft diverter and the chimney flue. Automatic dampers are usually designed to operate with electric contact mechanism, but frequently an arrangement is utilized which functions with mechanical fluid or gas pressure.

Physical Limitations

As it will usually be found that several boilers will meet the specifications, the final selection of the boiler may be influenced by other considerations, some of which are:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Space Limitations

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flues, and should be at least 3 ft greater than the length of the boiler firebox.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various

dampers, cleanouts and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

The velocity of flow through the outlets of low pressure steam heating boilers should not exceed 15 to 25 fps if fluctuation of the water line and undue entrainment of moisture are to be avoided. Steam or water outlet connections preferably should be the full size of the manufacturers' tapping and should extend vertically to the maximum height available above the boiler. For gravity circulating steam heating systems, it is recommended that a Hartford Loop, described in Chapter 16, be utilized in making the return connection.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. *I.P.S.*[9] if the gage is more than 5 ft from the boiler connection, and also in any case where the connection is less than ½ in. *I.P.S.*

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be direct connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

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[9]*A.S.M.E.* Code, Identification of Piping Systems.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapters 16 and 17 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer always should be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

The following precautions should be taken in all installations to prevent damage to the boiler:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.

4. Condensation must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source

of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When

pressure recedes close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.
2. All machined surfaces should be coated with oil or grease.
3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.
4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. A hot water system usually is left filled to the expansion tank.
5. The grates and ashpit should be cleaned.
6. Clean and repack the gage glass if necessary.
7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.
8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler

and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

PROBLEMS IN PRACTICE

1 ● What basic requirements of boiler design are to be accomplished with a combination boiler and oil burner unit?

Combination units vary widely but in general, the basic requirements of design depends upon a combustion chamber of proper design and arranged for the flame shape with adequate heating surface for the complete combustion of the fuel.

2 ● What is the normal rating range of each type of boiler?

a. Cast-iron boilers are rated at from 200 to 18,000 sq ft EDR.

b. Steel boilers are rated at from 300 to 50,000 sq ft EDR.

3 ● What factors contribute to economical fuel operation in low pressure boilers burning coal or oil?

a. Proper furnace volume for complete combustion.

b. Arrangement of heating surfaces in series to create a turbulent and scrubbing contact of gases against the convective surfaces.

c. Rapid internal water circulation which will remove steam bubbles from the water side of heating surfaces and allow other steam bubbles to be formed. Rapid disengagement of steam bubbles increases the steam generating efficiency of each unit area of heating surface, and thereby lowers flue gas temperatures.

4 ● What equipment is usually directly attached to a low pressure heating boiler?

For coal burning steam boilers: water column, water gage, tri-cocks, steam gage, lever pop safety valve, boiler damper regulator.

For coal burning hot water boilers: damper regulator, altitude gage, thermometer, relief valve.

For oil burning boilers, the damper regulators are omitted and the following additional equipment is usually attached: automatic water feeder, low water cutout, a pressure control, and a water temperature control. These are generally furnished by the oil burner manufacturer and do not come with the boiler.

5 ● What general precautions regarding the boiler should be taken to make sure a proposed heating installation will work properly?

a. Select the right size and type of boiler.

b. Be sure the combustion space is proper for the type of fuel burned.

c. Allow sufficient space around the boiler for cleaning.

d. Secure proper height and area of chimney and connecting breeching.

e. Clean the boiler thoroughly and provide surface blowoff connections and bottom blowoff connections for periodic cleaning after operation is begun.

f. See that the boiler heating surface is cleaned at regular periods.

g. Check flue gas temperatures and make a flue gas analysis at least once a month.

h. Secure information and advice from boiler manufacturer.

5 ● What is the average heat transmission rate in heating boilers in Btu per sq ft of heating surface per hour?

3500 for coal burning boilers; 4200 for oil burning boilers.

Chapter 14

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators, Convectors, Selection, Code Tests, Gravity-Indirect Heating Systems

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

All heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast iron. Catalogs showing the many designs and patterns available now include a junior size which is more compact than the standard unit.

Pipe Coils

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be

preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is the *Mb* or 1000 Btu.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet.* For steam service this is based on an emission of 240 Btu per hour per square foot.

TABLE 1. VARIATION IN DIMENSIONS AND CATALOG RATINGS OF 10-SECTION TUBULAR RADIATORS

No. of Tubes	3	4	5	6	7
Width of Radiator Inches	4.6–5.1	6.0–7.0	8.0–8.9	9.1–10.4	11.4–12.8
Length per Section Inches	2.5	2.5	2.5	2.5	2.5–3.0
HEIGHT WITH LEGS—INCHES	HEAT EMISSION—EQUIVALENT SQUARE FEET				
13–14	....	....	....	20	25.0–32.5
16–18	....	....	28.5	....	30.0–38.3
20–21	15.0–17.5	20.0–22.5	25.0–31.2	30	36.7–45.0
22–23	20.0–21.3	25	30.0–33.9	35	40.0–45.2
25–26	20.0–26.7	25.0–27.5	32.5–39.8	37.5–40.0	50.0–53.5
30–32	25.0–30.9	33.3–35.0	40.0–48.6	50	63.3–62.5
36–38	30.0–36.7	40.0–42.5	50.0–56.5	60	70.0–75.4

Output of Tubular Radiators

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation in design between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

Output of Wall Radiators

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests[1] show that this value will be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

[1]University of Illinois, *Engineering Experiment Station Bulletin* No. 223, p. 30.

Output of Pipe Coils

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

Effect of Paint

The prime coat of paint on a radiator has little effect on the heat output, but the finishing coat of paint does influence the radiation emission. Since this is a surface effect, there is no noticeable change in the convection loss. Thus, the larger the proportion of direct radiating surface, the greater will be the effect of painting on the radiation. Available tests are on old-style column type radiators which gave results shown in Table 3.

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

RADIATOR No.	FINISH	AREA SQ FT	COEFFICIENT OF HEAT TRANS. BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish	27	1.77	100.5
2	One coat of aluminum bronze	27	1.60	90.8
3	Gray paint dipped	27	1.78	101.1
4	One coat dull black Pecora paint	27	1.76	100.0

[a]Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 41).

Effect of Superheated Steam

Available research data indicate that there is probably a decrease in heat transfer rate for a radiator or gravity convector with superheated steam in comparison with saturated steam at the same temperature. The decrease is probably small for low temperatures of superheats and additional tests are necessary with varying degrees of superheat to establish accurate comparisons for all types of radiators and convectors[2].

[2]Tests of Radiators with Superheated Steam, by R. C. Carpenter (A.S.H.V.E. TRANSACTIONS, Vol. 7, 1901, p. 206).

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[3].

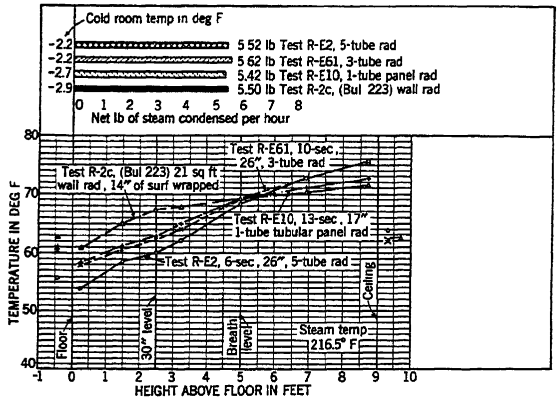

FIG. 1. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 60-IN. LEVEL

Note that the steam condensations are practically the same for all four radiators *when the same air temperature of 69 F is maintained at the 60-in. level.*

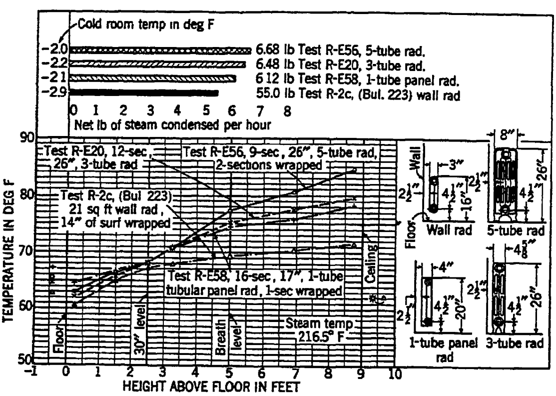

FIG. 2. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 30-IN. LEVEL

Note that the steam condensations are different for all four radiators *when the same air temperature of 68 F is maintained at the 30-in. level.*

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2[4]. For the four types of radiators shown, the following conclusions are given:

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[3]The Heating Effect of Radiators, by Dr. Charles Brabbeé (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33). The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

[4]Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 475).

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.
2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.
3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.
4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.
5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings[5].

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on[6]. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old style column type radiators.

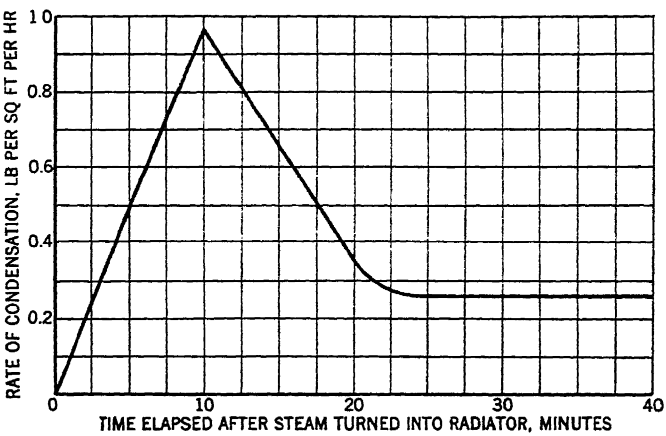


FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Recent investigations[7] indicate that in the design of the enclosure three things should be considered:

[5]Effect of Two Types of Cast-Iron Steam Radiators in Room Heating, by A. C. Willard and M. K. Fahnestock (*Heating, Piping and Air Conditioning*, March, 1930, p. 185).

[6]The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1937, p. 251).

[7]University of Illinois, *Engineering Experiment Station Bulletin* Nos. 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 77).

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the

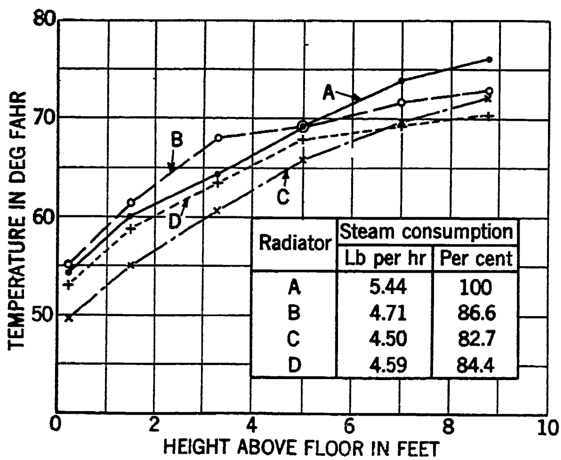


FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Practically all commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[8] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

CONVECTORS OR CONCEALED HEATERS

Although any standard radiator may be concealed in a cabinet or other enclosure so that the greater percentage of heat is conveyed to the

[8]University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p. 20.

room by convection thereby resulting in a form of gravity convector, generally better results are obtained with specially designed units which permit a free circulation of a larger volume of air at moderate temperatures. Since air stratifies according to temperature, moderate delivery temperatures at the outlet of the enclosure reduce the temperature differential between the floor and ceiling and accordingly accomplish the desired heating effect in the living zone.

Fig. 5 shows a typical built-in convector. The heating element consisting of a large percentage of fin surface is usually shallow in depth and

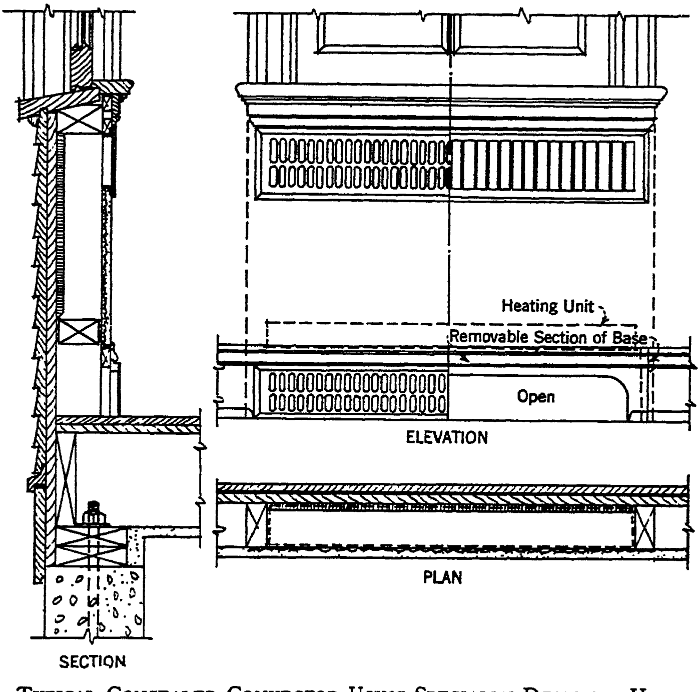


FIG. 5. TYPICAL CONCEALED CONVECTOR USING SPECIALLY DESIGNED HEATING UNIT

placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room which always lies at this level, is constantly being withdrawn and replaced by the warmer air. This air movement accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E. Standard[9] in the formulation of its ratings and has compiled a tentative

[9]A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367); (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237).

standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C.* Ratings (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. Code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association.*

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is important that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a variable of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu's. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation.

RADIATOR AND CONVECTOR SELECTION

Since the capacity of a radiator varies as the 1.3 power and a convector[10] as the 1.5 power of the temperature difference between the inside of radiator and surrounding air it is obvious that for other than 70 F room temperatures the heat emission will be other than 240 Btu per square foot of rating. Therefore in selecting the size of radiator or convector to be used it is necessary to correct for this difference. Table 4 shows factors by which radiation requirements, as determined by dividing heat load by 240, shall be multiplied to obtain proper radiator or convector sizes from published rating tables for room temperatures ranging between 50 and 80 F as well as for steam or water temperatures from 150 to 300 F. For other room and heating medium temperatures the factor is determined by the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3}$$

[10]Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 443).

For convectors:

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5}$$

where

C_s = correction factor.
t_s = steam temperature, degrees Fahrenheit.
t_r = room temperature, degrees Fahrenheit.
t_i = average inlet air temperature, degrees Fahrenheit.

TABLE 4. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTOR HEATERS[a]

Steam Press. Approx.		Steam or Water Temp. F	Factors for Direct Cast-Iron Radiators							Factors for Convectors						
			Room Temperature F							Inlet Air Temperature F						
Gage Vacuum In. Hg	Abs. Lb per Sq In.		80	75	70	65	60	55	50	80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In.																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a]To determine the heater size for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

To determine the heating capacity of a heater at other than standard conditions, divide the heating capacity at standard conditions by the proper factor from the above table.

CODE TEST FOR RADIATORS AND CONVECTORS

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam, 1932, and Hot Water, 1933).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power for radiators[11] and the 1.5 power for convectors[12] of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

Standard test conditions specify either a steam pressure of 1 lb gage (215 F), or hot water at 170 F and a room temperature of 70 F for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \tag{1}$$

[11]Loc. Cit. Note 9.
[12]Loc. Cit. Notes 9 and 10.

where

H_t = Btu per hour under test conditions.
W_s = condensation in pounds per hour.
h_{fg} = latent heat in Btu per pound.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} = \left(\frac{145}{T_s - T_r}\right)^{1.3} \tag{2}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.5} = \left(\frac{150}{T_s - T_i}\right)^{1.5} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test, degrees Fahrenheit.
T_r = room temperature during test, degrees Fahrenheit.
T_i = inlet air temperature during test, degrees Fahrenheit.
H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W(\theta_1 - \theta_2)\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test.
θ_1 = average temperature of inlet water, degrees Fahrenheit.
θ_2 = average temperature of outlet water, degrees Fahrenheit.
t = duration of test, seconds.

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} = \left(\frac{105}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} \tag{6}$$

It has been shown that when the exponent 1.5 is used the range of error is less than 3 per cent[13] for convectors.

GRAVITY-INDIRECT HEATING SYSTEMS[14]

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 6. The temperature and volume of the air leaving the register must be great

[13]Loc. Cit. Note 10.

[14]For further information on this subject see A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (edition of 1929) and *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. 1, second edition, 1929.

enough so that in cooling to room temperature the heat available will jus equal the heat loss during the same time. In cases where ventilation is a requirement, the air volume needed may become so large that the entering air temperature will be but slightly above the room temperature. To establish and maintain a constant heat flow, provision must be made fo removing the air in the room, after it has cooled to the desired room tem perature, by a system of vent flues or ducts. As the air flow is maintained

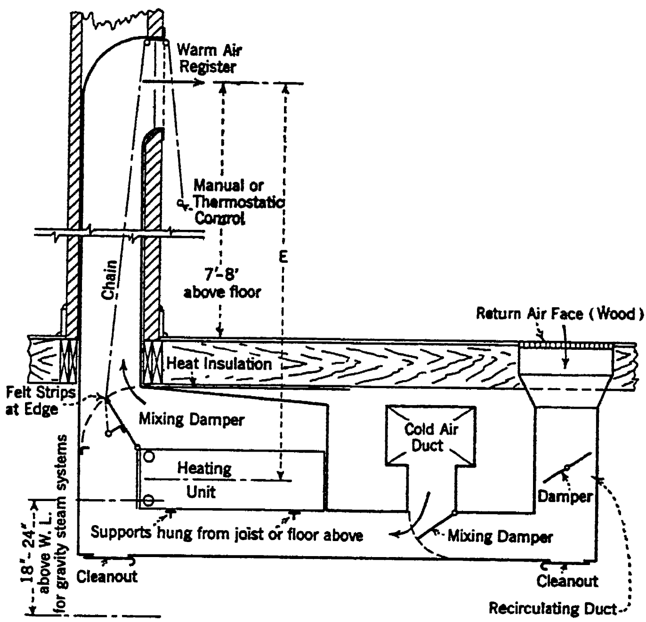


FIG. 6. GRAVITY-INDIRECT HEATING SYSTEM[a]

[a]See *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, especially the runs from the heating units to the base of the vertical warm air flues. Gravity-indirect arrangements, such as illustrated in Fig. 6, are not to be generally recommended for hot water systems unless the water temperature can be maintained at a reasonably high temperature and rapid circulation of the water can be had.

PROBLEMS IN PRACTICE

1 ● What is the effect on the heat output of a wall radiator when installed on the ceiling of a room?

Because the temperature differential is increased between the floor level and the ceiling when a wall radiator is placed near the ceiling, the heat output may be decreased from 5 to 10 per cent. Under such circumstances it becomes difficult to heat the living zone of a room satisfactorily.

2 ● What are the principal differences between a radiator and a convector?

A radiator is commonly thought of as a commercial heating unit having a maximum amount of direct heating surface, whereas a convector is a heating device in which the extended or secondary surface may be several times that of the prime surface and which is specially designed to utilize to the fullest extent the convection principal of heating. The radiator ordinarily has vertical tubular chambers for the heating medium but most convectors have horizontal tubular chambers to which fins are attached so as to form vertical flues for the passage of air. While radiators are either exposed, enclosed, or shielded, convectors are concealed by means of a tight-fitting enclosure. Radiators are commonly made of cast iron but convectors may be made of a combination of metals, such as copper and brass, or copper and aluminum, as well as entirely of cast iron.

3 ● How did the term heating effect come into use?

It has been found that a room requiring a radiator of a certain determined capacity could under certain conditions be properly heated, with less temperature gradient between floor and ceiling and with less steam condensation, by the same radiator or by one of a different design having the same commercially rated capacity. This resulted in the use of the term *heating effect* to apply to the useful heat output of a radiator, in the comfort zone of a room, as related to the total input to the radiator.

4 ● Is it necessary to make any allowance for the performance of a convector because it is enclosed?

No. The commercial ratings of convectors have been determined by testing the convectors in proper enclosures with grilles in place just as they should be installed for ordinary service.

5 ● On what basis are the capacities of convectors published?

Published ratings of convectors are expressed in equivalent square feet of direct cast iron radiation. Some manufacturers have increased their ratings by as much as 30 per cent to allow for a supposed improved heating effect. Tests indicate that the credit to be given heating effect is, in all cases, probably less than 10 per cent, and in many cases negligible.

6 ● How are fins of convectors attached to the tubes or prime surface?

Tubes or a solid core may be forced through piercings in the fins under pressure, or the tubes may be expanded into the holes through the fins. In addition a metallic bonding agent is sometimes used to insure permanent contact.

7 ● What is the procedure in selecting a convector when the required amount of radiation is known?

First the limiting factor or factors of the enclosure must be determined so the available size of the wall recess can be found. Manufacturers' catalogs show capacities of convectors of each standard length and depth with varying enclosure heights. From these capacity tables, the proper convector of the required capacity can be selected for the available wall recess. If all three dimensions of the wall recess are insufficient to accommodate a convector of the required capacity, the available height and length can be maintained, but greater depth can be obtained by using a partially recessed enclosure.

8 ● Given a room to be heated to 80 F with outside temperature at 0 F, assume the heat loss under these conditions to be 10,000 Btu per hour. Determine the size of the steam radiator to be installed.

A square foot of radiation is equivalent to a heat emission of 240 Btu per hour under standard conditions of steam at one pound gage pressure (215 F) and surrounding air at 70 F. With surrounding air at 80 F, the heat emission from a radiator will be less. Under these conditions, the heat emission will not be 240 Btu per square foot of catalog rating per hour, but 240 C_s.

$$C_s = \left(\frac{t_s - t_i}{215 - 70}\right)^{1.3} = \left(\frac{215 - 80}{215 - 70}\right)^{1.3} = 0.912,$$

and 240 C_s = 240 × 0.912 = 218.5 Btu. Therefore, the size of the radiator to be selected shall have a catalog rating of 10,000 divided by 218.5 or 45.8 sq ft.

Chapter 15

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent System, Gravity Two-Pipe Air-Vent System, One-Pipe Vapor System, Two-Pipe Vapor System, Atmospheric System, Vacuum System, Sub-Atmospheric System, Orifice System, Zone Control, Auxiliary Conditioning Unit, Condensation Return Pumps, Vacuum Pumps, Traps

THE essential features of the common type of steam heating systems are described in this chapter. They may be classified according to the piping arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. Information concerning the design and layout of steam heating systems will be found in Chapter 16.

GRAVITY AND MECHANICAL RETURN

In *gravity systems* the condensate is returned to the boiler by gravity due to the static head of water in the return mains. The elevation of the boiler water line must consequently be sufficiently below the lowest heating units and steam main and dry return mains to permit the return of condensate by gravity. The *water line difference*[1] must be sufficient to overcome the maximum pressure drop in the system and, when radiator and drip traps are used as in two-pipe vapor systems, the operating pressure of the boiler. The condensing return of the radiation will increase the required water line difference and is especially important where the radiation is a type having a high condensing rate. This applies only to closed circuit systems, where the condensation is returned to the boiler. If the condensation is wasted, no water line difference is required, but other conditions are introduced which warrant the use of an appropriate mechanical system in preference to wasting the condensate.

In *mechanical systems* the condensate flows to a receiver and is then forced into the boiler against the boiler pressure. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system, but the relative elevation of the boiler water line is unimportant in such cases

[1]The *water line difference* is the distance between the water line of the boiler and the level of the water in the dry or wet return main. (See Fig. 4.)

except that the head on the pump or trap discharge becomes greater as the height of the boiler water line above the trap or pump increases.

There are three general types of mechanical returns in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return pump. Further information on pumps and traps will be presented later in this chapter.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

In the gravity one-pipe air-vent system each radiator has but a single connection through which steam must enter and condensation must return in the opposite direction. Each radiator has an individual air valve.

Up-Feed Gravity One-Pipe Air-Vent System

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost of instal-

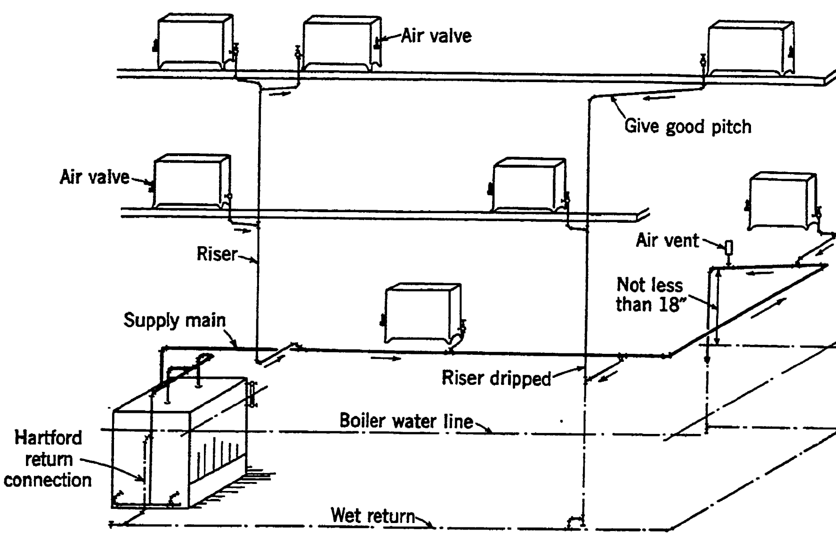


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

lation and its simplicity. Where the size of the system is moderate or large, it cannot be assumed that these systems will be lower in cost than two-pipe systems using steam traps. In some instances it has been found that the cost of one-pipe systems under these conditions is greater owing to the higher cost of labor and materials due to the larger pipe sizes. As will be seen from Fig. 1, the steam piping rises to a point as high as possible at the boiler and pitches downward from this location until the far end of the main or mains is reached. At the far ends drips are taken off at the low points of the steam mains, are water-sealed below the boiler water line, and then brought back to the boiler in a wet return. Single pipe risers

are branched off the main or mains to feed the radiators, the steam passing up the riser and the condensation flowing down it. The steam and condensation flow in opposite directions in the riser but after the condensation enters the steam main it flows in the same direction as the steam and is disposed of through the drip connection at the end of the main. In buildings of several stories, it is customary to drip the heel of each riser separately, whereas in one- or two-story buildings this is not necessary. Both types of branches and risers are shown in Fig. 1.

Rapid elimination of air and condensation from the steam piping is essential to the successful operation of this system. It is therefore desirable that the venting and dripping of the steam main in long runs be made at several intermediate points where the steam main may again be brought to a higher elevation.

It is desirable to install the air-vent valves on the steam main about a foot ahead of the drips, as is indicated in Fig. 1 to prevent possible damage to the mechanism of the air-vent valve by water, in case the valves are installed directly above the drips.

Horizontal branches to radiators and risers should be pitched at least ½ in. in 10 ft downward toward the riser or vertical pipe, and the hori-

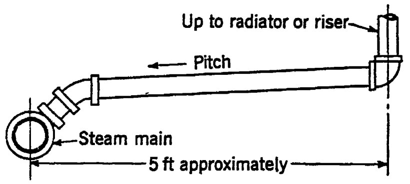


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

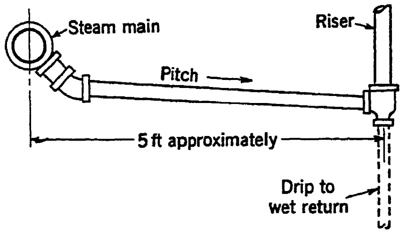


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

zontal branches from the steam main should be graded at least this amount toward the main, except where the heel of the riser is dripped, in which case the branch should pitch down toward the riser drip (Figs. 2 and 3). The return line, if wet, may be run without pitch or may be pitched in either direction, but if it is necessary to carry the return main overhead for any distance before dropping, the return should slope downward with the flow. It is desirable to install the wet return pipe with a pitch so that the system may be drained to prevent freezing in case the building remains unoccupied for a considerable length of time.

The radiator valves may be of the angle-globe or gate type. They should not be of the straight-globe type because the damming effect of the raised valve seat interferes with the flow of condensation through the valve. Graduated valves cannot be used, as the steam valves on this system must be fully open or closed to prevent the radiators filling with water. Air valves may be manual or automatic, with or without a check to prevent the re-entrance of expelled air. Usually the automatic type is installed. An objection to one-pipe steam systems is that the heat is all on or all off, with no intermediate position possible. However, intelligent

use of the on-and-off method of manual control gives reasonably satisfactory results. Improved systems and devices are now available which make it possible to obtain a modulating effect from one-pipe gravity heating systems.

It is important that the lowest points of the steam mains and heating units be kept sufficiently above the water line of the boiler to prevent

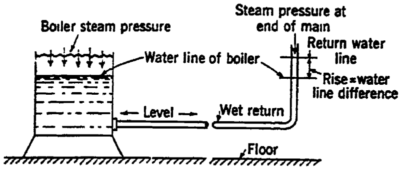


FIG. 4. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

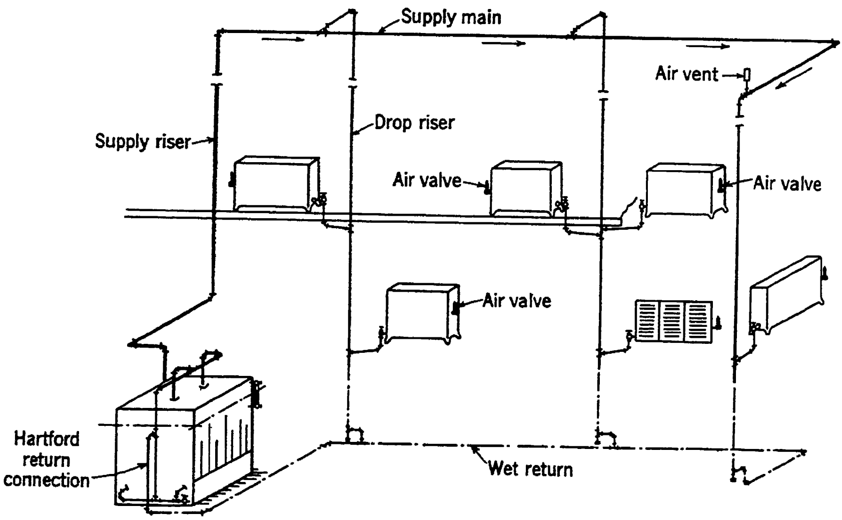


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

flooding. Usually 18 in. is sufficient but construction limitations frequently make shorter distances necessary. The distance may be checked in the following manner:

Referring to Fig. 4 it will be seen that the water in the wet return is really in an inverted siphon, or U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction of the steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise enough farther to produce a flow through the return into the boiler

(usually about 3 in. unless the pipes are small or full of sediment), and it will rise still farther if a check valve is installed in the return so as to obtain sufficient head to lift the tongue of the check (usually 4 in. will be necessary).

If a one-pipe steam system is designed, for example, for a total pressure drop of ⅛ lb, and utilizes an Underwriters' Loop[2] instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be ⅛ of 28 in., or 3½ in. Adding 3 in. to this for the flow through the return main and 6 in. as a factor of safety gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ lb, and with a check in the return, would require ½ of 28 in., or 14 in., for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

Down-Feed Gravity One-Pipe Air-Vent System

In the overhead down-feed gravity one-pipe air-vent system there is no change over the *up-feed system* in the radiators, the radiator valves, the air valves, or the radiator runouts as far back as the risers. Beyond this

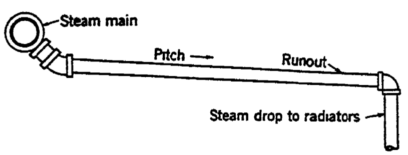


FIG. 6. STEAM RUNOUTS DRIPPING MAIN

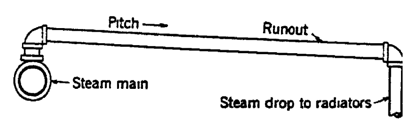


FIG. 7. STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

point there are basic differences. The steam is taken from the boiler and carried to the top of the building as near the boiler as possible (Fig. 5). If the run to the main riser is long, or if the riser extends several stories in order to reach the top, the bottom of the riser should be dripped into the wet return. The horizontal main is taken off the top of the riser and grades down from the riser toward all of the drops, each drop taking its share of the main condensation (Fig. 6), or all of the drops except the last may be taken from the top of the main (Fig. 7), the last drop being from the bottom and serving as a drain for the entire main. As the overhead main does not carry any condensation from the radiators it is immaterial which method is used. The air vent shown on the main just before the last drop (Fig. 5) may be placed at this point or it may be located at the bottom of the drop under the last radiator connection and sufficiently above the water line of the boiler to prevent flooding.

GRAVITY TWO-PIPE AIR-VENT SYSTEM

The gravity two-pipe system is now considered obsolete although many of these systems are still in use in older buildings. Separate supply and return mains and connections are required for each heating unit; air valves are installed on the heating units and mains; hand valves are installed on the returns.

[2]See discussion of piping details in Chapter 16.

Up-Feed Gravity Two-Pipe System

This system (Fig. 8) has a steam and a return connection to each radiator. The radiator valves for steam, return, and air are the same as those described for the gravity one-pipe air-vent system. The steam main is run and pitched in the same manner as in the one-pipe system, but the returns from each radiator are connected into a separate return line system which has its risers carried down and joined to a wet return line under the boiler water line level. Where the return has to be kept high to function as a dry return, it is advisable to connect the return risers to the dry return main through water seals about 36 in. deep, as

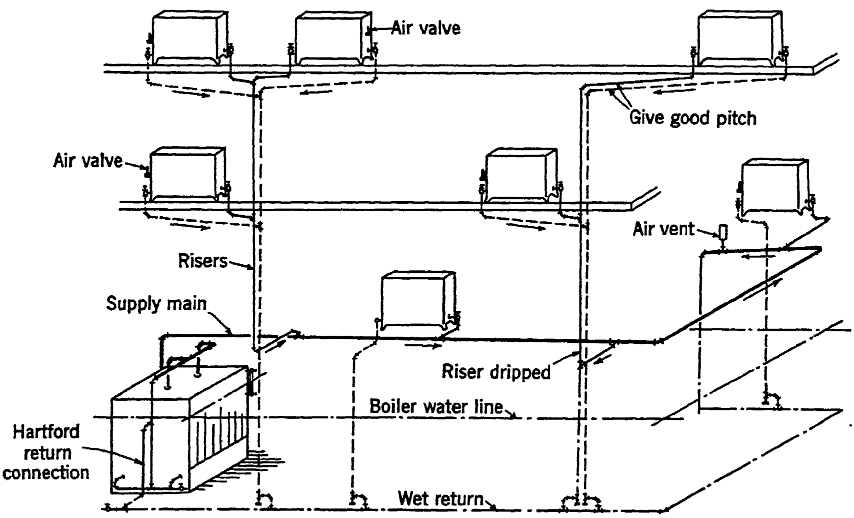


FIG. 8. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

shown in Fig. 9, to prevent steam from one riser entering another and closing the air valves on the nearest radiators.

Down-Feed Gravity Two-Pipe System

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged practically as in the down-feed one-pipe gravity system. The drips at the bottoms of the steam drops and the runouts to the radiators are similar to those shown in Fig. 8 for the up-feed gravity two-pipe system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

ONE-PIPE VAPOR SYSTEM

A vapor system is one which operates under pressures at or near atmospheric and which returns the condensation to the boiler by gravity. The piping arrangement of a one-pipe vapor system is similar to that of

the gravity one-pipe steam system; in fact, one-pipe gravity installations may readily be changed to one-pipe vapor systems by making a few simple alterations. The steam radiator valve is a plug cock which when opened gives a free and unobstructed passageway for water. The automatic air valve is of special design to permit the ready release of air from the radiator and to prevent the return of the air after it is expelled. The air valves on the main are a quick relief type, and the whole system is designed to operate on a few ounces of pressure.

TWO-PIPE VAPOR SYSTEM

Two-pipe vapor systems may be classified as (1) *closed systems* consisting of those which have a device to prevent the return of air after it is once expelled from the system, and which can operate at sub-atmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems* consisting of those

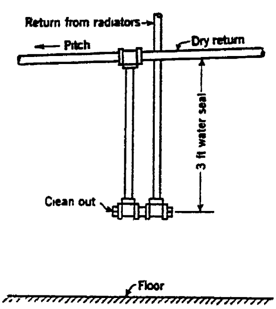


FIG. 9. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

which have the return line constantly open to the atmosphere without a check or other device to prevent the return of air, and which operate at a few ounces above atmospheric pressure. The open systems have the disadvantage of not holding heat when the rate of steam generation is diminishing.

Under the first classification the essentials are packless graduated valves on the radiators, thermostatic return traps on the returns, and traps on all drips unless they are water sealed. Such a system, illustrated in Fig. 10, should be equipped with an automatic return trap to prevent the water from backing out of the boiler. In this up-feed arrangement the supply piping is carried to a high point directly at the boiler and is graded down toward the end or ends of the supply main, each supply main being dripped at the end into the wet return or carried back to a point near the boiler where it drops down below the boiler water line and becomes a wet return. From this main, runouts are branched off to feed risers or radiators above, these being graded back toward the steam main if they are not dripped at the bottom of the riser, or toward the riser if the riser heel is dripped. Both conditions are illustrated in Figs. 2 and 3.

Return risers are connected to each radiator on its return end through

thermostatic traps. Their bottoms are connected to the return main through runouts which slope toward the main. The return main itself is sloped back toward the boiler if it is carried overhead; if run wet, the slope may be neglected, although it is desirable to slope the pipe so that the system may be drained. An air vent is installed at the point at which the return main drops below the water line. In the simplest cases this vent consists of a ¾-in. pipe with a check valve opening outward, but certain systems employ special patented forms of vent valves, designed to allow the air readily to pass out of the system and to prevent its return. A check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which usually is located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed

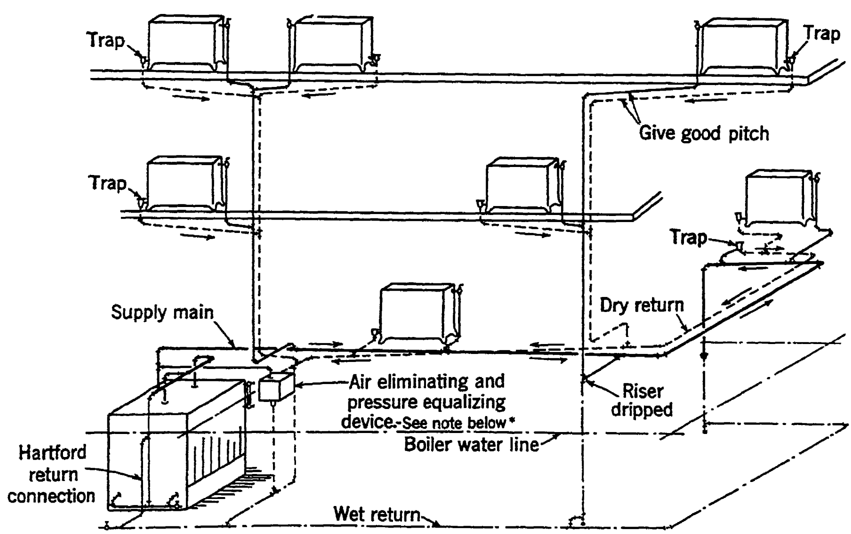


FIG. 10. TYPICAL UP-FEED VAPOR SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler (Fig. 11).

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts sloped toward the drops (Fig. 6). Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop. Another method of running the steam main, which is not considered as satisfactory but which is practical, is to take the branches off the top of

the main (Fig. 7) and to drip the end of the main through the last riser, as illustrated in the down-feed one-pipe system detail shown in Fig. 6. If this is done, the pipe drop at the end or ends of the mains should be enlarged one pipe size to provide capacity for this concentration of the main drip.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small

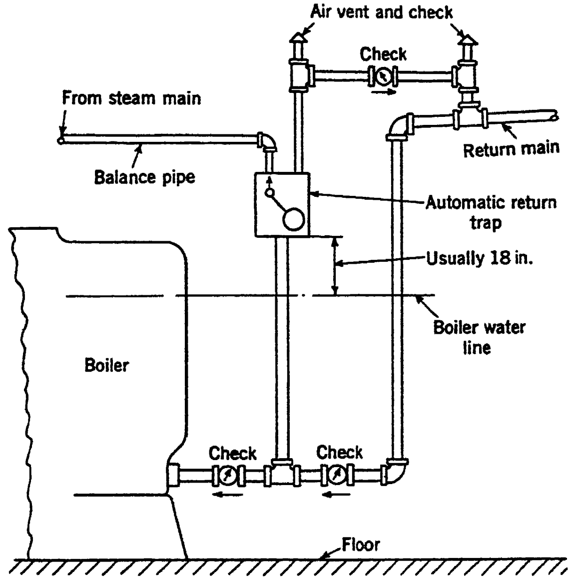


FIG. 11. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

and the normal size of drop required is 1 in. or less. The bottom of the steam drops should terminate with a dirt pocket above which a drip trap connection is located, as shown in Fig. 12. The returns on a down-feed vapor system are the same as on an up-feed system except that every steam drop must have a drip at the bottom connected either into the return through a trap or into a separate water-sealed drip line below the boiler water line, as illustrated in Fig. 10, in which case the thermostatic traps may be omitted. The runouts to the radiators and the radiator connections of the down-feed system are the same as those of the up-feed system already described.

ATMOSPHERIC SYSTEM

The distinguishing features of the atmospheric system are gravity

return to the boiler or to waste, graduated or ordinary radiator valves, no automatic air valves on the radiators, thermostatic traps on the radiator returns, and the venting of all air from the system by means of pipes open to the atmosphere. The returns are open to the atmosphere at all times, usually by extending the return risers to the top of the building where they are either connected together in groups and carried through the roof or extended through the roof individually. Atmospheric systems, either up-feed or down-feed, are often used where the condensation is not returned to the boiler, as in heating systems supplied by high pressure steam through pressure-reducing valves at locations far from the boilers. The returns may be delivered back to the boiler, if desired, by condensation return pumps which are vented to the atmosphere. The return lines in such systems are simply gravity waste lines in which the condensation flows entirely by gravity and is not aided by any pressure difference

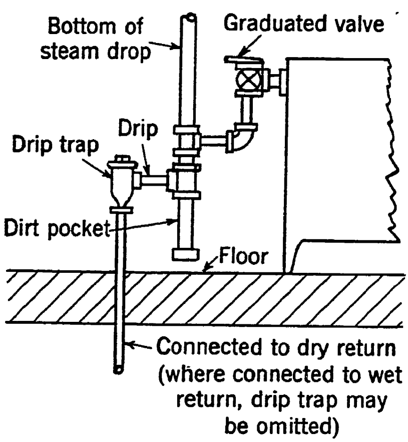


FIG. 12. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

Atmospheric systems contemplate maintaining a practically constant pressure in the steam pipe and atmospheric pressure in the return pipe. When graduated steam valves are provided, they enable the occupant of a room to vary the flow area to the radiator so as to obtain a greater or lesser heating effect.

The steam side may be run as that for either up-feed or down-feed two-pipe vapor systems, as the conditions require, and the radiator connections are the same as for vapor systems in that they have graduated valves on the radiator supply ends and thermostatic traps on the radiator return ends. All drips from the supply main and the steam side of the system must pass through thermostatic drip traps before entering the return system where only atmospheric pressure exists. Fig. 13 illustrates a typical scheme of piping used on atmospheric systems. Such systems do not maintain heat in the radiators under declining fires. As the steam supply diminishes, air from the atmosphere re-enters through the open vent pipe retarding the inflow of steam and cooling the radiator.

VACUUM SYSTEM

In the vacuum system, a vacuum is maintained in the return line practically at all times but no vacuum is carried on the steam side, and the usual accessories include graduated valves on the radiator supply and thermostatic traps on the radiator return. The air is expelled from the system by a vacuum pump and all drips must pass through thermostatic traps before connecting to the return side of the system.

These systems are often fed from high pressure steam mains through pressure-reducing valves but they may be fed direct from a low-pressure steam heating boiler as shown in Fig. 14, in which a typical up-feed vacuum system is illustrated. The supply main slopes down in the direction of flow; the runouts pitch down toward the riser if the riser is dripped (Fig. 3) or up toward the riser if the riser is not dripped (Fig. 2); both conditions are indicated in Fig. 14. The matter of dripping the risers depends largely on the height of the riser and the judgment of the designer. Ordinarily risers less than three stories high are not dripped and those more than four stories high are dripped, but there is no set rule for this. When risers are dripped the runouts from the steam main may be taken from the bottom if desired and each runout then serves as a drip for the main.

The risers are carried up to the highest radiator connection and are connected to the radiator through runouts sloping back toward the riser. The radiators usually have graduated valves on the supply end, although this is not absolutely necessary. Angle-globe valves and gate valves may be used where graduated manual control is not desirable. The return valves must be of the thermostatic type which will pass air and water but which will close against the passage of steam.

The return risers are connected in the basement into a common return line, which slopes downward toward the vacuum pump. The vacuum pump discharges the air from the system and pumps the water back to the boiler, or other receiver, which may be a feed-water tank or a hot well. It is essential on these systems that no connection from the supply side to the return side be made at any point except through a trap.

While the best practice demands a return flowing to the vacuum pump in an uninterrupted downward slope, in some cases limitations make it necessary to drop the return below the level of the vacuum pump inlet before the pump can be reached. In such event one of the advantages of the vacuum system is that the return can be raised by the suction of the vacuum pump to a considerable height, depending on the amount of vacuum maintained, by means of a lift fitting inserted in the return. Best practice dictates that the lift should be limited to a single lift connection at the entrance to the vacuum pump and preferably that an accumulator tank or receiver with float control be used at the low point of the return main at the entrance to the vacuum pump. When the lift is considerable, several lift fittings should be used in steps (Fig. 15), more successful operation being obtained by this method than when the lift is made in one step. If the lift occurs close to the vacuum pump, a special arrangement is used as shown in Fig. 16. It is desirable that means be provided for draining manually the low point of the lift fittings to eliminate from the return piping all water in danger of freezing in case

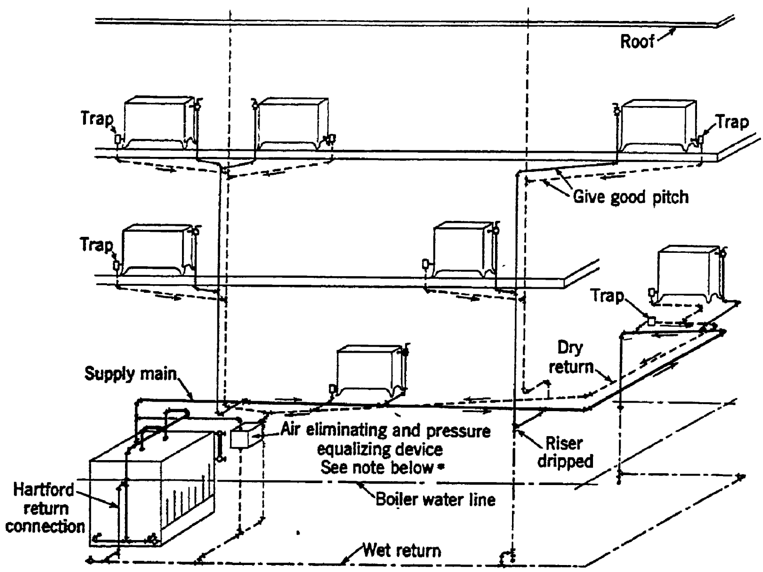


FIG. 13. TYPICAL ATMOSPHERIC SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air eliminatior

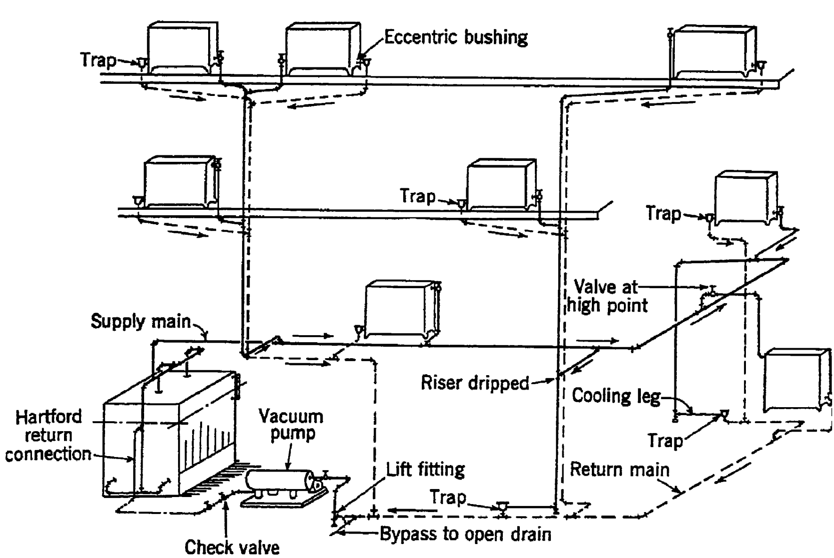


FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

the system is shut down for a considerable length of time. Lifts for draining condensate from ends of or rises in steam mains should be avoided to secure the greatest economy of operation and noiselessness.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that

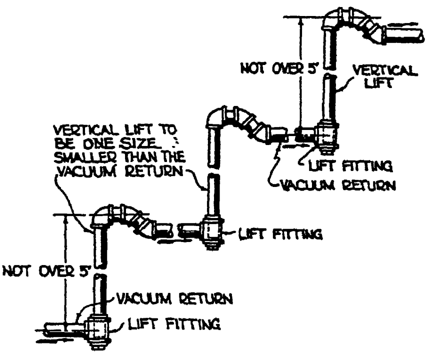


FIG. 15. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

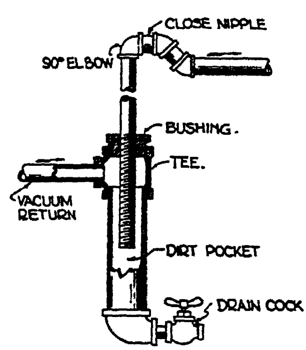


FIG. 16. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

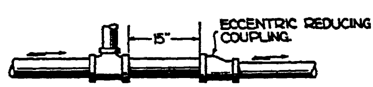


FIG. 17. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main (Fig. 6) so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 17). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 18.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems, but in contrast provide temperature control by variation of the heat output from the radiators both by varying the pressure at which steam is circulated in the radiation and the amount of steam. The steam supply is continuous at varying rates. A vacuum pump capable of operating at high partial

vacua is preferable since the higher the vacuum the greater is the accuracy in the distribution of steam through the system, particularly in mild weather. A pump capable of producing up to 25 in. of vacuum on the system is used in such cases. A controller is placed on the pump so that the vacuum or absolute pressure carried in the returns can be maintained at a certain amount below that existing in the line to insure circulation.

The traps are designed to operate in high vacuum. It is apparent that this system differs from the ordinary vacuum system by having a vacuum on both sides of the system, instead of only on the return side, in order to secure control of the heat emission from the radiators and thus to control

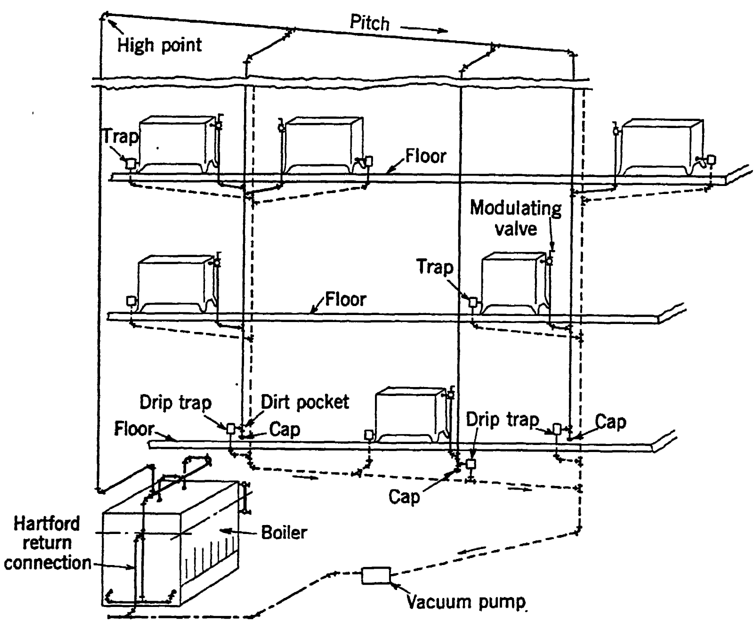


FIG. 18. TYPICAL DOWN-FEED VACUUM SYSTEM

the temperature in the building. These systems permit the heat output from the steam mains and risers to be diminished as the weather becomes milder, thus giving control to this portion of a heating system. The decrease in condensation in the piping as the temperature of the steam is reduced under a vacuum is a measure of the saving in heat loss from piping resulting from steam circulation at sub-atmospheric pressures as compared with circulation at sub-atmospheric pressure. The system can be operated in the same manner as the ordinary vacuum system when desired.

In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, steam pressure exists in the steam main and radiators only during the most severe weather, while under average winter temperatures the steam is under a partial vacuum which in mild

weather may reach as high as 25 in. after which further reduction in heat output is obtained by partially filling the radiation with steam.

This vacuum is partially self-induced by the condensation of the steam in the system due to the supply of steam being furnished through the control which admits it, and it being proportioned to balance the existing heat loss. To convert an ordinary vacuum return line system to a sub-atmospheric system, a control valve is inserted on the steam main near the boiler or the boiler is automatically controlled. The steam supply to each radiator is provided with a flow proportioning device, such as an orifice, a high-vacuum pump is substituted for the ordinary type and is supplied with a pressure-difference control, and traps are placed on the radiators and drips which will operate satisfactorily at any pressure from 5 lb gage to 26 in. of vacuum.

The control valve is either a special pressure-reducing valve which may be controlled manually, or a control valve or combustion equipment which may be operated thermostatically from points selected in the building. The vacuum pump regulator is simply a diaphragm so arranged that, when the vacuum in the return line is insufficient to hold the desired difference in pressure between the steam and return sides of the system, the vacuum pump is automatically started and the vacuum increased to the necessary amount. The actual pressure difference maintained between the two sides of the system is only enough to secure adequate circulation and is often about 2 in. of mercury. This fixed pressure difference between the supply and return sides of the system results in practically constant circulation under all pressure conditions.

In order to distribute the steam equally when the system is being warmed up and also to reduce the amount of steam delivered to the radiators on mild days, orifice plates are used in the graduated radiator control valves. A definite, nearly constant, relation exists between the supply and return pressure differential at various points throughout the system which promotes proportionate steam distribution between the various radiators. The heat emitted from the radiators in mild weather and under conditions of high vacuum is not only reduced in proportion to the difference in the steam temperature between that for 2 lb gage and for 25 in. of vacuum but it is reduced still further by a reduction in the amount of steam which can pass through the orifice when the steam is expanded due to the vacuum. This renders possible the control of heat emission from the radiators to a point not indicated entirely by the difference in steam temperatures, but far beyond it.

Sub-atmospheric operation has advantages even where individual thermostatic radiator control is installed. By operating the system with steam temperatures in parallel with the outside temperature requirements, a large part of the load is removed from the temperature control system, it makes fewer operations and the radiator follows an even temperature without fluctuating from extreme hot to extreme cold.

The high-vacuum pumps on this system are equipped with receivers having float control so that the pump can be placed on a receiver-return-pump basis at night if desired so no high vacuum will be carried. One radical difference between this system and the ordinary vacuum system is that no lifts can be made in the return line, except at the vacuum pump.

The returns must grade downward constantly and uninterruptedly from the radiator return outlet to the inlet on the high-vacuum pump receiver.

No attempt should be made to heat service water on this system unless the steam line for water heating is taken off the boiler header back of the heating system control valve, and then only when 2 lb or more will be carried on the boiler at all times. Sub-atmospheric systems are proprietary.

ORIFICE SYSTEM

Orifice systems of steam heating may have piping arrangements identical with vacuum systems but some of these systems omit both the radiator thermostatic traps and the vacuum pump in cases where the returns are wasted to a sewer or delivered to some type of receiver in which no back pressure exists. The principle on which they operate is embodied in the well-known fact that an orifice will deliver varying velocities when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side no further increase in velocity will be obtained.

As a result, if an orifice is so designed in size as to exactly fill a radiator with steam at 2-lb gage on one side and ¼-lb gage on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 90 \text{ per cent}$$

Should the steam pressure be dropped to ¼-lb gage, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be seen that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, it is simply a question of dropping this main pressure provided the supply pipe pressures be controlled sufficiently closely, so as to fill any desired portion of the radiator down to the point where the main pressure equals the back pressure in the radiator, at which time no steam will be supplied at all. If orifices throughout a system are designed on a similar basis, all radiators will heat proportionately to the steam pressure within the limits for which the orifices are designed.

Some systems use orifices not only in radiator inlets but also at different points on the main, thus balancing the system to a greater extent. For example, the system may be designed for a particularly long run involving an initial pressure of 3-lb gage on the main and 2 lb at the end of the main, but each branch from the main may have an orifice for reducing the pressure at it to 2-lb gage. This is particularly useful for branches near the boiler where the drop in the main has not yet been produced.

Orifice systems using a vacuum pump operate successfully with the ordinary low vacuum type of pump producing 8 to 10 in. of vacuum. They are controlled by various means to regulate the steam pressure. One method is by a thermostat located on the roof to govern the steam pressure by a combination of outside and inside temperatures; another, useful on systems without traps and vacuum pumps, controls the steam pressure manually from temperature indication stations in the building,

or automatically by a thermostatically-controlled pressure reduction valve or draft regulator on the boiler; with oil or gas firing, the on-and-off control or a boiler pressure control may be used.

ZONE CONTROL

Certain portions of a building may require more heat at times than others but if the whole building is on one general control, such as would occur with a single piping system with an on-and-off control or with the sub-atmospheric or the orifice systems, it would be necessary to supply sufficient heat to accommodate the coldest portion of the building even though some sections would be overheated. By separation of a building into zones each with its own piping system, each zone of the building may be controlled separately.

The sides of the building with different exposures should be considered first, because of the varying effects of the wind and sun. With the prevailing winter winds from the northwest, a simple zoning would place the north and west sides of the building on one system and the south and east sides on another. If the building is large enough to justify the expenditure, a better arrangement would be to place all north walls on one zone, all west walls on a second, all east walls on a third, and all south walls on a fourth.

In case of high buildings, the lowest 8 or 10 stories may be well protected from wind by surrounding buildings, the next 10 stories may have moderate exposure, and above this there may be an unobstructed exposure to gales. On still days the heat demands vertically will vary little, but on windy days there will be a marked difference in the heat requirements for the different horizontal sections. In addition, the *chimney effect* caused by the difference in density between the warm air on the inside of a building and the colder air on the outside will give an air movement which will require zoning to correct. Where such conditions are encountered, the building should be divided horizontally as well as vertically. An arrangement of this character would give 12 zones: namely, north, east, south, and west lower zones; similar middle zones; and similar top zones. Each zone should constitute an individual and separate system of piping with its own supply steam valve (controlled by thermostats in its respective zone) and with its own return or vacuum pump, if one is used. Certain interior areas, such as basements, light well walls and other locations where sun and wind do not affect the conditions, should be placed in still another zone if the most economical results are to be secured.

Zoning has advantages even where individual thermostatic radiator control is installed whether this be of pneumatic, electric, or the self-contained radiator valve type. By operating each zone to supply heat in parallel with its outside temperature and wind fluctuations, a large part of the load is taken off the thermostatic controls; they operate less frequently and the radiators follow a more even temperature instead of fluctuating from extreme hot to extreme cold.

Sub-atmospheric, orifice, and zone control systems, generally are proprietary. Sub-atmospheric systems may be zoned to care for exposure, occupancy and stack effect.

AUXILIARY CONDITIONING UNIT

In connection with a residential steam or hot water system using radiator or convector heating a unit as shown in Fig. 19, is available to supplement the old or new system. The unit is arranged in a sheet metal enclosure with a filter, circulating fan, means for adding moisture to the air, heating or tempering coil and generally provisions are made for the addition of a cooling coil in case summer air circulation is desired. The unit is frequently located on the ceiling of the basement and is connected with one or more supply and return air ducts in the various rooms. In some cases, provisions are made for the introduction of a portion of the outside air to the system and dampers are included to adjust the desired air quantities.

The heating coil of the unit may be connected to a steam or hot water boiler system and is adaptable for operation with a one-pipe, two-pipe or vacuum system. The cooling coil may be connected to a source of

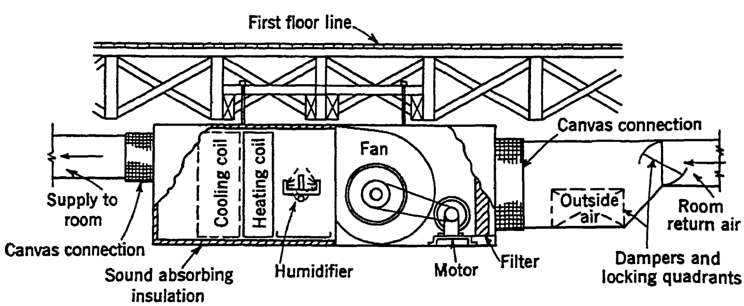


FIG. 19. RESIDENTIAL CONDITIONING UNIT

refrigeration, or in some cases city water is circulated through the coil when 58 F or lower temperature water is available. The amount of moisture released is adjustable depending upon the degree of humidification desired. The complete unit may be adapted to various automatic control arrangements to satisfy the comfort demands of the occupants.

CONDENSATION RETURN PUMPS

Whenever the conditions of a heating system are such that the returns from the radiation can not gravitate freely to the boiler, they must be returned by some mechanical means such as a condensation pump or a return trap.

The most generally accepted condensation pump unit for low pressure heating systems consists of a motor driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw and reciprocating pumps with steam turbine or motor drive, and direct acting steam reciprocating pumps.

Fig. 20 illustrates a typical installation of a motor driven automatic condensation unit. It will be noted that the returns flow by gravity to

the vented receiver. As the receiver is filled, the float mechanism operates either a pilot or an across-the-line switch to start the pump, and upon emptying the tank disconnects the power and stops it. The pump may be used to deliver the condensate direct to the boiler, to a feedwater heater or to raise the water to any higher elevation or pressure than that of the return line.

A useful application, for instance, is to use a small condensation unit to handle a remote section of radiation that otherwise would be difficult to grade to the main return.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly

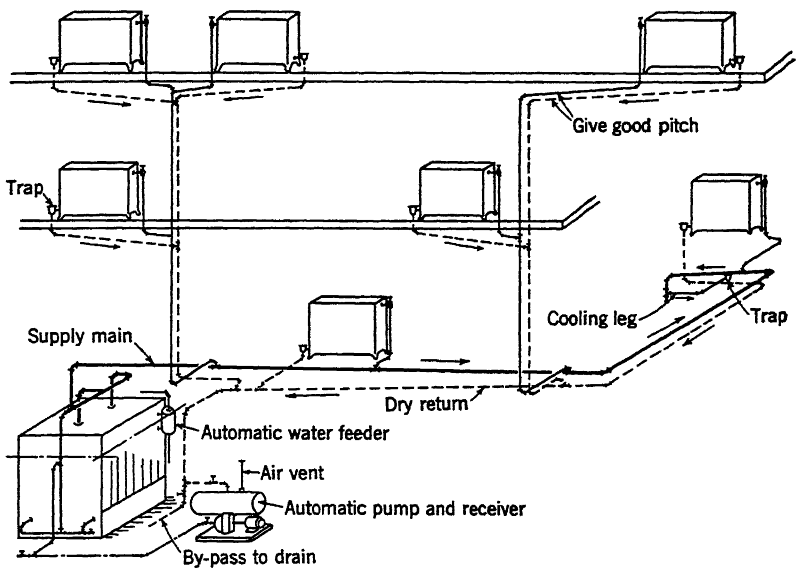


FIG. 20. TYPICAL INSTALLATION USING CONDENSATION PUMP

to the boiler and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute and the pump generally has a delivery rate of 3 to 4 times the normal flow.

VACUUM HEATING PUMPS

On vacuum or sub-atmospheric systems where the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and non-condensable gases to atmosphere and to return the condensate to the boiler. Direct acting steam driven reciprocating vacuum pumps are sometimes used where high pressure steam is available or where the exhaust steam from the pump can be utilized, but in general these have been replaced by the automatic motor driven return line vacuum heating

pump especially developed for this service. The usual unit consists of a compact assembly of air and water removal units driven by one motor and furnished complete with receiver, separating tank and full automatic controls mounted as an integrated unit on one base.

Practically all of such return line vacuum heating pumps make use of the returned condensate to operate either as a liquid piston or as a jet to withdraw the air, and in many cases the condensate, from the return line. Such hydraulic evacuating devices may be classified as:

a. Water ring centrifugal displacement pumps.
b. Water piston pumps.
c. Stationary water ejector pumps.
d. Rotary water ejector pumps.

The evacuating element is generally combined with a centrifugal water impeller for the delivery of the condensate to the boiler or feedwater heater.

The assembled units may be further grouped under two general classifications:

a. Those which perform the function of air separation under atmospheric pressure.
b. Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification remove both the air and condensate from the returns by means of the hydraulic evacuator and deliver both to a separating tank under atmospheric pressure. From this tank the air and non-condensible vapors are vented to atmosphere while the condensate is removed and delivered to the boiler by means of the built-in boiler feed pump impeller.

In the second classification, the air and condensate are first separated under vacuum by means of the receiver which is directly connected to the returns. The hydraulic evacuator withdraws only the air and non-condensible vapors from the top of the receiver and delivers them to atmosphere. The built-in condensate pump impeller removes the condensate from the bottom of the receiver and delivers it direct to the boiler or feedwater heater.

Under special conditions such as returning the condensate to a high pressure boiler or the furnishing of large air removal units for high vacuum systems, it is customary to supply separate motor driven air and water pumps. Steam turbine drive is also frequently used where high pressure steam is available. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 lb drop across the turbine required for its operation.

For rating purposes[3] vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated for maintaining 5½ in. mercury vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above 5½ in.

The vacuum that may be maintained on a system depends upon the

[3]A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps. (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33).

relationship of the operating air capacity of the hydraulic evacuator at the vacuum and temperature of the returns to the air leakage rate into the system. It is particularly essential on high vacuum installations that the system be tight and that steam be prevented from entering the return lines through leaky traps, high pressure drips, etc.

Vacuum Pump Controls

In the ordinary vacuum system the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and which cuts out when the vacuum has been increased to the highest point. This is done largely to eliminate the constant starting and stopping of the vacuum pump which would occur if the vacuum were maintained constant. In addition to this control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum in the system. A selector switch is usually provided to allow operation at night as a condensation pump only, also to give continuous operation if desired.

There are several variations to the above control, especially as concerns the control of the vacuum maintained on the system. This may be accomplished by some form of coordinating control which maintains the vacuum of the return system in a pre-determined definite or varying relationship to the system supply pressure.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either power or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump and at an elevation sufficiently high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensation is used for average vacuums and systems. However, as in the case of return line vacuum heating pumps, the displacement is largely dependent upon the tightness of the system, the efficiency of the traps and the vacuum that is desired to be maintained.

TRAPS

Traps are used for draining the condensate from radiators, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. The usual functions of a trap are to allow the passage of condensate and to prevent the

passage of steam. In addition to these functions, traps are frequently required to allow the passage of air as well as condensate. Traps are also required to allow the passage of air and to prevent the passage of either water or steam, or both.

In addition, traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may be classified as to function as *separating* and *return* or *lifting* traps. Traps may be classified according to the principle of operation as (1) float, (2) bucket, (3) thermostatic, (4) tilting, or (5) float and thermostatic traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Float and thermostatic traps have both a thermostatic element to release air and a float element to release the water.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used. but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting

traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

Thermostatic traps are generally used for draining radiators and heaters, except for very large capacities where bucket, float or blast-type thermostatic traps are used. Thermostatic traps for this service usually pass both condensate and air and in the case of float and upright bucket traps the air is usually relieved through an auxiliary thermostatic trap in a by-pass around the main trap. Sometimes this auxiliary air trap is an integral part of the trap. Such traps are termed float and thermostatic traps.

Blast-type thermostatic traps are sometimes used on vacuum heating systems for connecting old one- or two-pipe gravity systems in parallel with vacuum return line systems, in which case the blast-type thermostatic traps should not be provided with auxiliary air by-pass, as the

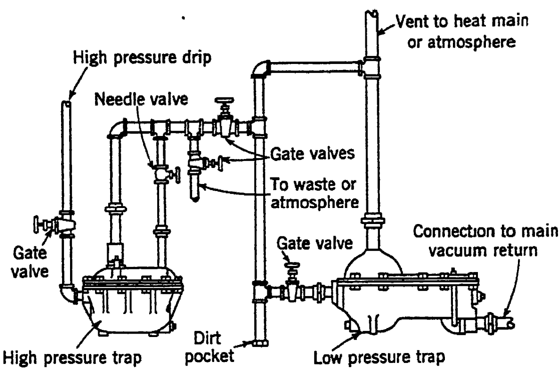


FIG. 21. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW-PRESSURE TRAP

action of this will allow the vacuum to draw air into the old system through its air valves, especially when the steam is wholly or partially cut off. The air from the returns of such old systems should be relieved just ahead of the traps by means of quick-venting automatic air valves, preferably of the non-return type, especially if the other air valves on the old system are non-return valves.

Return traps used for discharging to a higher or a lower pressure are provided with two or three valves operated by the action of the trap. In the case of the two-valve return traps, one valve closes a steam inlet and the other valve opens a vent outlet while the trap is filling, and as soon as the trap dumps, the first valve opens the steam inlet and the second valve closes the vent outlet, while the trap discharges. In this type of trap there must be a swinging check-valve on each side of the trap, in addition to the usual by-pass, to prevent the pressure in the trap, while discharging, from backing up through the inlet and the pressure in the discharge line from backing up into the trap while it is filling. This

type of trap will blow steam out through the vent while filling, if the pressure on the inlet side is sufficient, and should not be used, therefore, with such pressures unless the vent is properly piped back into the return

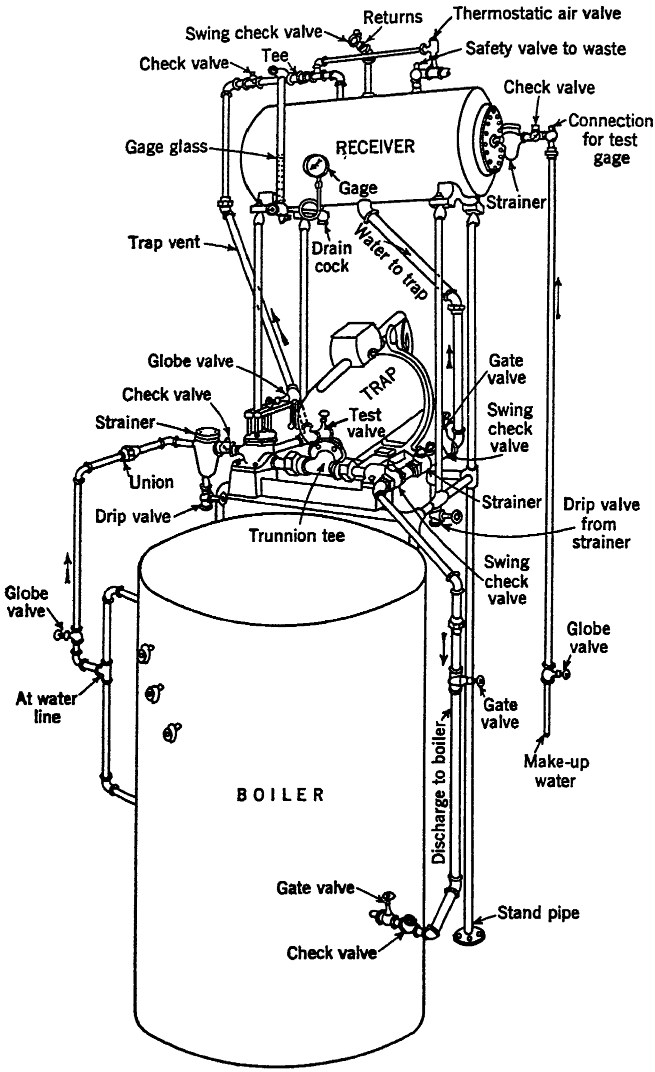


FIG. 22. RETURN TRAP AND RECEIVER FOR AUTOMATIC BOILER FEED

to a feed water heater, a condenser or a perforated pipe in the bottom of the receiver to which the trap discharges in such a way as to prevent the escape of the steam that comes in with the condensate and passes through the vent. In the three-valve traps of this type there is an extra

valve for closing the discharge while the trap is filling with condensate. High pressure traps should not discharge directly into a vacuum return because of the vapor formed by the re-evaporation of a part of the hot condensation. Fig. 21 shows a method which may be used for disposing of the greater part of the vapor of re-evaporation. An expansion chamber often is installed between the high- and low-pressure traps.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler. Fig. 22 shows a direct return tilting trap and receiver properly connected for automatically feeding a boiler from a system of returns delivering the condensate to the receiver.

PROBLEMS IN PRACTICE

1 ● What is meant by water line difference in a gravity steam heating system?

The water line difference is the distance between the level of the water in the dry or wet return and the boiler water line. This difference is equivalent to the pressure required to overcome the maximum drop in the system and the operating pressure of the boiler.

2 ● How many types of common mechanical returns are there and what are they?

Three: (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum pump.

3 ● In the ordinary vacuum system of steam heating, where does the vacuum usually exist?

On the return side of the system only, between the radiator trap and the vacuum pump. If the radiator supply valve is closed off, the vacuum may extend back through the radiator as far as the supply valve; if an inadequate supply of steam is furnished to the system, some vacuum may be developed in the steam main, but neither of these can be termed *normal operation.*

4 ● What is the distinction between the open and the closed vapor systems?

The open vapor system has the return line always open to the atmosphere, while the closed vapor system has an automatic device on the air vent so that air once expelled from the system through the vent cannot re-enter via this route.

5 ● On a vacuum system, what device must be placed on all drips before they enter the vacuum return line?

A thermostatic drip trap or occasionally, where large volumes of condensation are to be handled, a float trap, or combination float and thermostatic trap.

6 ● How does the sub-atmospheric system differ in operation from the ordinary vacuum system?

The ordinary vacuum system has pressure in the steam line, and a vacuum produced by the vacuum pump in the return line, usually varying between 5 and 10 in. of mercury. The sub-atmospheric system may have either a vacuum or pressure on the steam and return lines according to the weather conditions, but a constant difference in pressure is maintained between the lines regardless of what vacuum may be carried. The vacuum, which is generally produced jointly by condensation and the exhausting action of the pump, in the system under conditions of throttled steam supply, will run much higher than in the ordinary vacuum system, and as high as 25 in. of mercury in the radiators.

7 ● What is generally understood by zoning in building steam heating systems?

Zoning is a term applied to the placing of certain sections of a building on a single temperature control instead of having either individual room control or a single temperature control governing the whole building. Zones may be horizontal, such as a single story, a basement, or an attic, or vertical such as the north side, or the west side.

8 ● Why does the water line in the far end of a wet return in a gravity steam system rise higher than the water line in the boiler?

The friction of the steam flowing through the steam main from the boiler to the far end of the system and the pressure reduction resulting from the condensing action of the radiators causes a drop in steam pressure at the point where the wet return is connected; consequently, the steam pressure on top of the water in the wet return is less than the steam pressure on top of the water in the boiler, so the water in the end of the wet return rises until a balanced condition is set up.

9 ● On gravity one-pipe systems as indicated in Fig. 1 and Fig. 3, why is the drip on the steam runout connected to wet return?

Because if it were connected to dry return, the pressure drops to two different points would not necessarily be the same and the system would short circuit.

10 ● What is the function of the automatic return trap?

To insure the return of condensate to the boiler when the operating condition is such that the boiler pressure exceeds the static head on the returns.

11 ● What advantage is there to an air valve with a check to prevent the reentrance of expelled air?

A system equipped with such valves builds up a vacuum and holds the heat longer. With proper controls on the boiler, lower radiator temperatures can be maintained in mild weather, giving better plant efficiency.

12 ● What are the essentials of a two-pipe closed vapor system?

Packless graduated valves on radiators; thermostatic return traps on return and drips; an automatic return trap to prevent water from backing out of the boiler.

13 ● Why must the automatic return trap on two-pipe vapor systems be about 18 in. above the boiler water line?

That height is necessary to overcome water line difference owing to pressure drop and friction in pipe and fittings.

Chapter 16

PIPING FOR STEAM HEATING SYSTEMS

Flow of Steam in Pipes, Pipe Sizes, Tables for Pipe Sizing, One-Pipe Gravity Air Vent Systems, Two-Pipe Gravity Air Vent Systems, Two-Pipe Vapor Systems, Vacuum Systems, Atmospheric Systems, Sub-Atmospheric Systems, Orifice Systems, High Pressure Steam, Expansion in Steam and Return Lines, Piping Connections and Details, Boiler Connections, Hartford Return Connection

THE design of a steam heating system should be considered under four headings, namely, (1) the details of the heating units, (2) the arrangement of the general piping scheme, (3) the details of connections, and (4) the sizing of the lines. Items 1 and 2 are covered in Chapters 14 and 15, respectively, while this chapter considers the two latter items.

The functions of piping are to supply the heating units with steam and to remove the condensation. In some systems both the air and condensation are removed from the heating units by the return piping. To accomplish this effectively, the distribution of the steam should be efficient and equitable, without noise, and the returns should be as short as possible. When air is handled its escape should be facilitated to the utmost since an air-bound system will not heat properly. Condensation takes place in a steam system not only in the heating units, but throughout the piping system as well, and the returns also condense any steam or vapor that may be contained. At the same time part of the condensation may flash back into steam when the vacuum or pressure in the return is considerably below the steam pressure.

It is essential that steam piping systems not only distribute steam at full load but also at partial loads, as the average winter demand is less than half of the demand in most severe outside temperatures. Furthermore, in heating up rapidly the load on the steam main may exceed the maximum operating load even in extreme weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature. This may require more heat than would be emitted from the system itself after it once is thoroughly heated.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship of flow of dry steam or steam with a small amount of water has been

established by Babcock in formula 1.

$$P = 0.0000000367 \left(1 + \frac{3.6}{d}\right) \frac{W^2L}{Dd^5} \tag{1}$$

or

$$W = 5220 \sqrt{\frac{PDd^5}{\left(1 + \frac{3.6}{d}\right) L}} \tag{2}$$

where

P = loss in pressure, pounds per square inch.
d = inside diameter of pipe, inches.
L = length of pipe, feet.
D = weight of 1 cu ft of steam.
W = weight of steam flowing per hour, pounds.

Example 1. How much steam will flow per hour through 100 ft of 2-in. pipe if the initial pressure is 1.3 lb per square inch and the pressure drop is 1 oz?

Solution. $P = \frac{1}{16} = 0.0625$ lb; $d = 2.067$ in. (Table 1, Chapter 18); $L = 100$ ft; $D = 0.04038$ lb (Table 8, Chapter 1). Substituting these values in Formula 2:

$$W = 5220 \sqrt{\frac{0.0625 \times 0.04038 \times 2.067^5}{\left(1 + \frac{3.6}{2.067}\right) 100}} = 97.2 \text{ lb per hour.}$$

Formula 2 does not allow for entrained water in low-pressure steam, condensation in pipe, and roughness in commercial pipe as found in practice.

The latent heat of steam (h_{fg}) at atmospheric pressure (Table 8, Chapter 1) is 970.2 Btu per pound. Inasmuch as the heat emission of an equivalent square foot of heating surface (radiation) is 240 Btu, 1 lb of steam at this pressure will supply $\frac{970.2}{240}$ or 4.04 sq ft of equivalent heating surface. This figure is usually taken as 4 even. In Example 1, the weight of steam flowing per hour would therefore supply 4 × 97.2 or 388.8 sq ft of equivalent heating surface.

PIPE SIZES

The determination of pipe sizes for steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. Unusual conditions in the building to be heated.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmos-

pheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; (4) there is sufficient difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, and the dry return, when considered in relation to the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 1 and 2 for vertical risers and in Table 3 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1-in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 4). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 5). The third is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the pipe on which Table 4 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

TABLE 1. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet Per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0.68	45	10.961	11.3
1¼	17.6	0.66	98	23,765	24.5
1½	20.0	0.66	152	36,860	38.0
2	23.0	0.57	288	69,840	72.0
2½	26.0	0.54	464	112,520	116.0
3	29.0	0.48	799	193,600	199.8
3½	31.0	0.44	1144	277,000	286.0
4	32.0	0.39	1520	368,000	380.0

INSTRUCTIONS FOR USING TABLE 1

1. Capacities given in Table 1 should never be exceeded on one-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5).

TABLE 2. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	-----	40	9550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 2

1. The capacities given in this table should never be exceeded on two-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5.)

TABLE 3. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 6 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLE 4. PER CENT DIFFERENCE IN CAPACITY FOR CARRYING STEAM AND CONDENSATE DUE TO VARIATION OF PIPE SIZE AND SMOOTHNESS[a]

	MAXIMUM CONDENSATION, LB PER HOUR			
Size of pipe	¾ In.	1 In.	1¼ In.	1½ In.
Minimum	14.00	24.89	45.42	70.50
Maximum	15.20	30.08	52.08	82.00
Per cent variation	8.6	20.8	14.7	16.3

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 5. EFFECT OF REAMING ENTRANCE TO ONE-INCH ONE-PIPE RISERS[a]

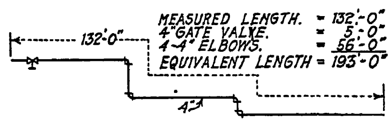
	MAXIMUM CAPACITY OF RISER	PER CENT DECREASE
Reamed entrances	24.7 lb per hour	0.0
Rounded entrances	23.9 lb per hour	3.2
Squared entrances	22.2 lb per hour	10.1
Three wheel cutter	19.2 lb per hour	22.2
Single wheel cutter	17.6 lb per hour	28.7

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 6. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

Size of Pipe Inches	St'd. Elbow	Side Outlet Tee	Gate Valve	Globe Valve	Angle Valve
	Length in Feet to be Added to Run				
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING[1]

Factors determining the size of a steam pipe and its allowable limit of capacity are as follows:

1. Pipe condensate flowing with steam.
2. Pipe condensate flowing against steam.
3. Pipe and radiator condensate flowing with steam.
4. Pipe and radiator condensate flowing against steam.

It is apparent that (3) and (4) are practically limited to one-pipe systems while (1) and (2) cover all other systems.

Tables 7 and 8, worked out for determining pipe sizes, have their columns lettered continuously, Columns *A* through *L* being in Table 7, and *M* through *EE* in Table 8. In the following text, reference made to columns will be by letter. The tables are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation[2] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

[1]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[2]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression *square foot of equivalent direct radiation*, *EDR*, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A.S.H.V.E.

Table 7 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve, and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 8 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 7. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

TABLE 7. STEAM PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

	CAPACITIES OF STEAM MAINS AND RISERS								SPECIAL CAPACITIES FOR ONE-PIPE SYSTEMS ONLY		
	DIRECTION OF CONDENSATION FLOW IN PIPE LINE										
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only				
PIPE SIZE IN.	1/32 lb or ½ Oz Drop	1/24 lb or ⅔ Oz Drop	1/16 lb or 1 Oz Drop	⅛ lb or 2 Oz Drop	¼ lb or 4 Oz Drop	½ lb or 8 Oz Drop	Vertical	Horizontal	Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Runouts
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
¾	-------	-------	30	-------	-------	-------	30	-------	25	-------	-------
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	-------	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	-------	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	-------	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	-------	1,110
5	3,214	3,712	4,546	6,429	9,092	12,858	-------	3,150	-------	-------	2,180
6	5,276	6,094	7,462	10,553	14,924	21,105	-------	-------	-------	-------	-------
8	10,983	12,682	15,533	21,967	31,066	43,934	-------	-------	-------	-------	-------
10	20,043	23,144	28,345	40,085	56,689	80,171	-------	-------	-------	-------	-------
12	32,168	37,145	45,492	64,336	90,985	128,672	-------	-------	-------	-------	-------
16	60,506	69,671	84,849	121,012	169,698	242,024	-------	-------	-------	-------	-------
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Un-dripped Run-outs	Up-Feed Risers	Radiator Con-nections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 7.

TABLE 8. RETURN PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pipe Size Inches	CAPACITY OF RETURN MAINS AND RISERS																	
	1/32 Lb or 1/2 Oz Drop per 100 Ft			1/24 Lb or 2/3 Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			1/8 Lb or 2 Oz Drop per 100 Ft			1/4 Lb or 4 Oz Drop per 100 Ft			1/2 Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	MAINS																	
3/4	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,977
1 1/4	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	......	......	3,390
1 1/2	1,350	822	......	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2 1/2	4,700	3,040	......	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,925
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	......	......	30,230
3 1/2	11,000	7,880	......	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	......	......	62,180
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100
	RISERS																	
3/4	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,977
1	......	450	......	......	450	976	......	450	1,200	......	450	1,704	......	450	2,400	......	......	3,390
1 1/4	......	990	......	......	990	1,547	......	990	1,900	......	990	2,696	......	990	3,800	......	......	5,370
1 1/2	......	1,500	......	......	1,500	3,256	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,453	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,925
2 1/2	......	......	......	......	......	8,710	......	......	10,700	......	......	15,910	......	......	21,400	......	......	30,230
3	......	......	......	......	......	13,020	......	......	16,000	......	......	22,710	......	......	32,000	......	......	45,200
3 1/2	......	......	......	......	......	17,910	......	......	22,000	......	......	31,220	......	......	44,000	......	......	62,180
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
5	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution. It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ lb, while if the total drop were $\frac{1}{2}$ lb, the drop per 100 ft would be $\frac{1}{20}$ lb. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

One-pipe gravity air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-lb drop (Column *D*).
2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.
3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.
4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.
5. *For the radiator valve size and the stub connection*, use Column *K*.
6. For the *dry return main*, use Column *U*.
7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{24}$-lb drop is being used, the steam main and dripped runouts would be sized from Column *C*; radiator runouts and undripped riser runouts from Column *L*; up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{24}$ lb); the dry return from Column *R*; and the wet return from Column *Q*.

With a $\frac{1}{32}$-lb drop the sizing would be the same as for $\frac{1}{24}$ lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Example 3. Size the one-pipe gravity steam system shown in Fig. 1 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ lb the drop per 100 ft will be slightly less than $\frac{1}{16}$ lb. It would be well in this case to use $\frac{1}{24}$ lb, and this would result in the theoretical sizes indicated in Table 9. These theo-

retical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.
3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.
4. Supply mains, branches to risers, or risers, should be dripped where necessary.

TABLE 9. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 1

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED (SQ FT)	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to radiators	....	100	2	2
Branches to radiators	....	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	3½	3½
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main	*f* to *k*	600	1¼	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

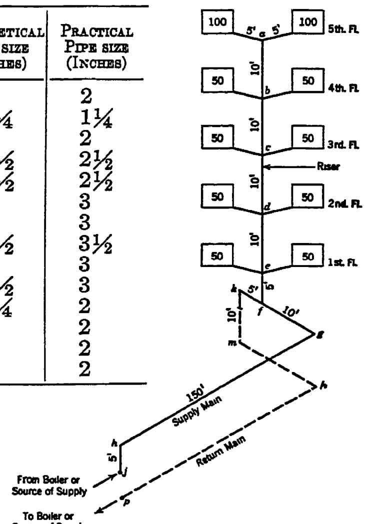


FIG. 1. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of ½ lb the drop per 100 ft would be $\frac{\frac{1}{2}}{8}$ or $\frac{1}{16}$ lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of ¼ lb is desired, the drop per 100 ft would be $\frac{1}{32}$ lb

and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be ⅛ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is 1/64 lb or 1/32 lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 8 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire, and (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ lb to ¼ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ lb divided by 4, or 1/32 lb. In this case the steam mains would be sized from Column *B*; the radiator and

undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than 1/4 in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general it is not desirable to have a supply main smaller than 2 in., and when the supply main is 3 in. or over at the boiler or pressure reducing valve it should not be less than 2 1/2 in. at the far end.

4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return. The drip for a vapor system may be connected into the dry return through a thermostatic drip trap.

VACUUM SYSTEMS

Vacuum systems are usually employed in large installations and have total drops varying from 1/4 to 1/2 lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of 1/2 lb divided by 12, or 1/24 lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of 1/24 lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed 1/8-lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.

2. Pitch of mains should not be less than 1/4 in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in. When the supply main is 3 in. or over, at the boiler or pressure reducing valve, it should not be less than 2 1/2 in. at the far end.

5. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 15 under *Up-Feed Vacuum Systems*.

ATMOSPHERIC SYSTEMS

The sizing of the supply and return piping on atmospheric systems is practically identical with the sizing used for vacuum systems and the same notes apply, except that no lift can be made in the return line.

SUB-ATMOSPHERIC SYSTEMS

Any properly pitched, correctly sized vacuum system without a lift except at the vacuum pump may be used as a sub-atmospheric system when the proper equipment is substituted for the ordinary vacuum pump, traps, and controls. On new systems manufacturers usually recommend a drop on the steam line of between ¼ and ½ lb for the total run, and suggest adding 25 ft to the total equivalent length of run to insure that the steam gets through to the last radiator.

The same notes apply to these systems as for vacuum systems, except that no lifts can be made in the returns.

ORIFICE SYSTEMS

The orifice systems can be operated with any piping system suitable for vacuum operation, according to experienced designers. Because these systems vary considerably in detail, it is advisable to consult the manufacturer of the particular system contemplated for recommendations.

The same notes apply to these systems as to vacuum systems, except that lifts cannot be made in the returns of orifice systems if a vacuum pump is used.

HIGH PRESSURE STEAM

When steam heating systems are supplied with steam from a high pressure plant, one or more pressure-reducing valves are used to bring the pressure down to that required by the heating system. It has been considered good practice to make the pressure reductions in steps not to exceed 50 lb in each case. For example, in reducing from 100-lb gage to 2-lb gage, two pressure reducing valves would be used, the first reducing the pressure from 100-lb gage to 50 lb and the second reducing the pressure from 50-lb gage to 2-lb gage. Valves are available that will reduce 100 lb in one step, and it is questionable whether two valves are now required for initial pressures of 150 lb or less.

The pressure-reducing valve, or pressure-regulator as it is sometimes termed, has ratings which vary 200 to 400 per cent. Some of these ratings are based on arbitrary steam velocities through the valve of 5,000 to 10,000 fpm and it is assumed that the valve when wide open has the same capacity as the pipe on the inlet opening of the valve. At times it is considered desirable to keep the steam velocity in the high pressure section of the piping and the low pressure section constant. The velocity through the valve port is obviously a function of the pressure drop across the valve. It is well known that steam flowing through an orifice increases its velocity until the pressure on the outlet side is reduced to 58 per cent of the absolute pressure on the inlet side, and that with further reduction of pressure on the outlet side little change in velocity will be obtained. As practically all pressure-reducing valves used for steam heating work

lower the steam pressure to less than 58 per cent of the inlet pressures, only the maximum velocity through such valves need be considered. If it is assumed that the valve, when fully open, has an area equal to that of the inlet pipe size, that the steam is flowing into a pressure less than 58 per cent of the initial pressure, that the orifice efficiency is approximately 70 per cent, and that 20 per cent more is allowed for a factor of safety, then the pressure reducing valves will have the working capacities shown in Table 10. If the valve, when fully open, does not give an orifice area equal to that of the pipe on the inlet side, then the capacities will be proportional to the percentage of opening secured, taking the pipe area as 100 per cent. More frequently, difficulty is encountered from the use of pressure reducing valves which are too large in size instead of being

TABLE 10. CAPACITIES OF PRESSURE-REDUCING VALVES
(100-LB GAGE DOWN TO ANY PRESSURE—52 LB OR LESS)

Inlet Nominal Pipe Diameter (Inches)	Pounds Steam per Hour at 100-Lb Gage	Equivalent Direct Radiation Sq Ft at ¼ Lb	Equivalent Direct Radiation Sq Ft at ⅛ Lb
½	866	3,464	2,598
¾	1,576	6,304	4,728
1	2,459	9,836	7,377
1¼	4,263	17,052	12,689
1½	5,808	23,232	17,424
2	9,564	38,256	28,692
2½	13,623	54,492	40,869
3	21,041	84,104	63,123
3½	28,213	112,852	84,039
4	36,285	145,140	108,855
5	56,971	227,884	170,913
6	82,336	329,344	247,008

Formula:

$$\frac{A \times V \times 3600 \times .50}{144 \times 3.88} = \text{pounds per hour passed by orifice.}$$

where

A = area of inlet pipe, square inches.
V = velocity of steam through orifice (approximately 870 fps).
50 = 70 per cent efficiency of orifice less 20 per cent for factor of safety.
144 = square inches in 1 sq ft.
3600 = seconds in one hour.
3.88 = cubic feet per pound at 100-lb gage.

too small. Where valves are large in size, the valve tends to work close to the seat, causing it to cut out in a relatively short time, as well as being noisy in operation.

Most exact regulation of pressure on steam heating systems is secured from diaphragm-operated valves controlled by a pilot line from the low pressure pipe, taken off the low pressure main at least 15 ft from the reducing valve. The reducing valves operating on the proportional-reduction principle will give a variation of steam pressure on the low pressure side if the initial pressure varies between considerable limits. The so-called dead-end valve is used for reduced pressures where the line has not sufficient condensing capacity at all times to condense the leakage that might occur with the ordinary valve. Single-disc valves do not give as close regulation as double-disc valves, but the single disc is preferable where dead-end valves are necessary, such as on short runs to thermo-

statically controlled hot water heaters, central fan heating units and unit heaters.

The correct installation (Fig. 2) of a pressure-reducing valve includes a pressure-reducing valve with a gate valve on each side, a by-pass controlled by a globe valve, a pressure gage on the low pressure side, and a safety valve on the low pressure main at some point, usually within a reasonable distance of the pressure-reducing valve. Pressure-reducing valves should have expanded outlets for sizes greater than 2 in. Where the steam main is of still larger diameter than the expanded outlet, and in cases where straight valves are used, an increaser is placed close against the outlet of the valve to reduce the velocity immediately after passing through the valve. Strainers are recommended on the inlets of all pressure-reducing valves. A pressure gage may be located on the high-pressure line near the valve if desired.

Owing to the large variation in steam demand on the average heating system, it is generally advisable to use two pressure-reducing valves con-

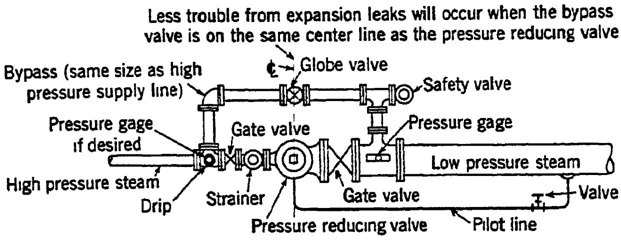


FIG. 2. TYPICAL PRESSURE-REDUCING VALVE INSTALLATION

nected in parallel. One valve should be large enough for the maximum load and the other should have a diameter approximately half that of the first. The smaller valve can be used most of the time, for it will give much better regulation than the larger one on light or normal loads.

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

EXPANSION IN STEAM AND RETURN LINES

Because all steam and return lines expand and contract with changes in temperature, provision should be made for such movement. The expansion in steam supply pipes is normally taken at 1¼ to 1½ in. per 100 ft and in return lines at one-half or two-thirds of this amount. It may be calculated accurately if the temperature rise and fall can be determined with reasonable certainty (Chapter 18). The temperature at the time of erection often has a greater expansion effect on piping than the temperature in the building after it has been put into service.

Expansion may be taken care of by any, or all, of three different methods, namely, (1) the spring in the pipe including offsets and expansion bends, (2) the turning of the pipe on its threads and swing joints, and (3) the use of expansion joints.

By the first scheme, which is the most popular method where space permits, the pipe is offset, or *broken*, around rooms or corners, and is hung so that the spring in the pipe at right angles to the expansion movement is sufficient to absorb the expansion. If conditions do not lend themselves to this treatment, regular expansion bends of the *U* or offset type may be used. In tight places such as pipe tunnels the expansion joint is preferable. See additional material on pipe expansion bends in Chapter 18.

On riser runouts and radiator runouts the swing joint is used almost without exception. On high vertical risers the pipes may be reversed every five to ten stories; that is, the supply is carried over to the adjacent return riser location and the return riser is run over to the former supply riser location, thus making horizontal offsets in each line. Corrugated copper expansion joints also are used on risers but must be made accessible in case future replacement becomes necessary.

PIPING CONNECTIONS AND DETAILS

Piping connections may be classified into two groups: *first,* those suitable for any system of steam heating; *second,* those devised for certain systems which cannot be satisfactorily applied to any other type. There are also various details that apply to piping on the steam side which cannot be used on the returns. An installation that is designed and sized correctly and installed with care may be rendered defective by the use of improper connections, such as runouts that do not allow for expansion, thermostatic traps unprotected from scale, pressure-reducing valves without strainers, and lack of drips at required points.

BOILER CONNECTIONS

Supply

Boiler headers and connections have the largest sizes of pipe used in a system. Cast-iron, horizontal-type, low pressure heating boilers usually have several tapped outlets in the top, the manufacturers recommending their use in order to reduce the velocity of the steam in the vertical uptakes from the boiler and to permit entrained water to return to the boiler instead of being carried over into the steam main where it must be cared for by dripping. Steel heating boilers usually are equipped with only one steam outlet but many engineers believe that better results are obtained by specifying that such boilers have two. The second outlet, usually located 3 or 4 ft back of the regular one, reduces the velocity 50 per cent in the steam uptake.

Fig. 3 shows a type of boiler connection that was used for many years and one with which some boilers are now piped. The uptakes are carried as high as possible, turned horizontally and run out to the side of the boiler and then are connected together into the main boiler runout which drops into the top of the boiler header through a boiler stop valve. No drips are provided on this type of runout except a very small one which is sometimes installed on the boiler side of the stop valve. Fig. 4 shows a

type of boiler connection which is regarded as superior to that shown in Fig. 3 and which is the type illustrated in the system diagrams in Chapter 15. This type is similar to that shown in Fig. 3 except that the horizontal branches from the uptakes are connected into the main boiler runout, and the steam is carried toward the rear of the boiler. The branch to the building or boiler header is taken off *behind* the last horizontal boiler connection. At the rear end of this main runout, a large size drip, or balance pipe, is dropped down into the boiler return, or into the top of the Hartford Loop, which is described in a following paragraph. As a result, any water carried over from the boiler follows the direction of steam flow

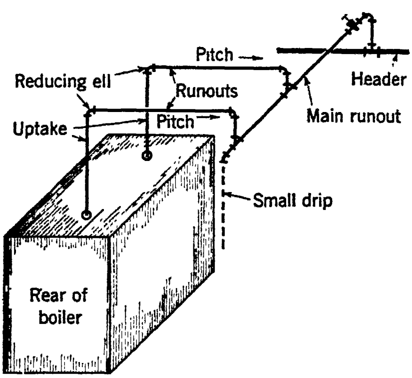


FIG. 3. OLD STYLE STANDARD BOILER CONNECTIONS

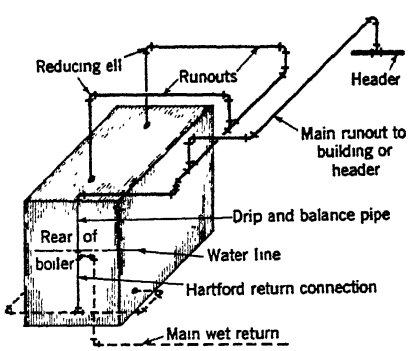


FIG. 4. APPROVED METHOD OF BOILER CONNECTIONS

toward the rear and is discharged into the rear drip, or balance pipe, without being carried over into the system.

Return

Cast-iron boilers are generally provided with return tappings on both sides, but steel boilers often are equipped with only one return tapping. A boiler with side return tappings will usually have a more effective circulation if both tappings are used. Check valves generally should not be used on the return connection to steam heating boilers from one and two pipe gravity systems because they are not always dependable inasmuch as a small piece of scale or dirt lodged on the seat will hold the tongue open and make the check useless. These valves also offer a certain amount of resistance to the returns coming back to the boiler, and in gravity systems will raise the water line in the far end of the wet return several inches[3]. However, if check valves are omitted and the steam pressure is raised with the boiler steam valve closed, the water in the boiler will be blown out into the return system with the accompanying danger of boiler damage. These objections are largely overcome with the Hartford return connection.

[3]See method of calculating height above water line for gravity one-pipe systems in Chapter 15

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford Connection, or the Underwriters Loop, is recommended.

Fig. 5 shows this connection for a two-boiler installation. For a single boiler installation the connection is made as is indicated for one boiler. The essential features of construction of a Hartford Loop connection are: (1) A direct connection (made without valves) between the steam side of

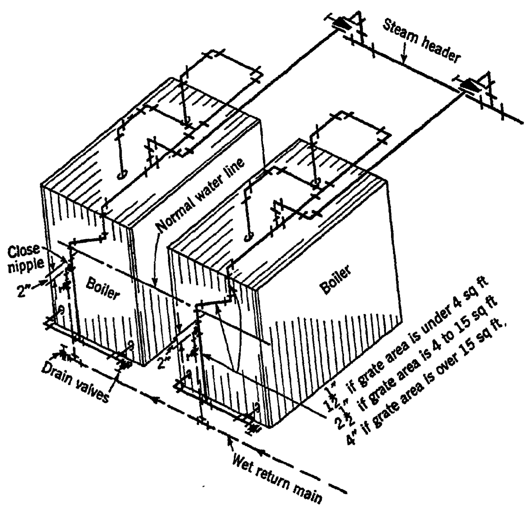


FIG. 5. THE HARTFORD RETURN CONNECTION

the boiler and the return side of the boiler, and (2) a close nipple connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return balance connection.

Sizing Boiler Connections

Little authentic information is available on the sizing of boiler runouts and steam headers. Although many engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the

boiler and the horizontal main or runout is compensated for by the use of reducing ells (Figs. 3 and 4).

The following example illustrates the sizing of the boiler connections shown in Fig. 6.

Example 4. Determine the size of boiler steam header and connections (Fig. 6) if there are three boilers, two to carry 50 per cent of the load each, and the third to be used as a spare. The steam mains are based on 1/8-lb drop per 100 sq ft of equivalent direct radiation (EDR).

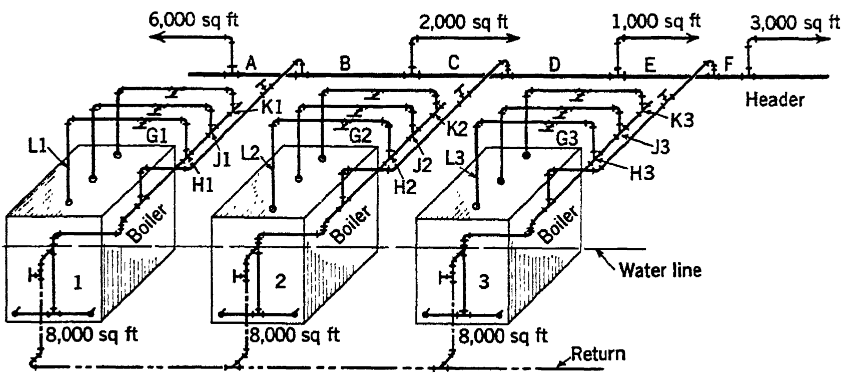


FIG. 6. BOILER STEAM HEADER AND CONNECTIONS

Solution:

Size of Boiler Header

WHEN OPERATING ON BOILERS	LOAD ON VARIOUS PORTIONS OF HEADER						MAXIMUM LOAD
	A	*B*	*C*	*D*	*E*	*F*	
Nos. 1 and 2	6000	0	2000	4000	3000	3000	6000
Nos. 2 and 3	6000	6000	8000	2000	3000	3000	8000
Nos. 3 and 1	6000	0	2000	2000	3000	3000	6000
Max. Load	6000	6000	8000	4000	3000	3000	8000

8000 sq ft @ ⅛ lb per 100 ft = 6 in. main. (See Table 7.)

Size of Boiler Runouts

The three runouts

G_1, G_2, G_3 = $\frac{8000}{3}$ = 2667 sq ft each @ ⅛ lb per 100 ft = 4 in. pipe.

H_1, H_2, H_3 = 2667 sq ft each @ ⅛ lb per 100 ft = 4 in. pipe[4] (See Table 7).

J_1, J_2, J_3 = 5333 sq ft each @ ⅛ lb per 100 ft = 5 in. pipe[4] (See Table 7).

K_1, K_2, K_3 = 8000 sq ft each @ ⅛ lb per 100 ft = 6 in. pipe[4] (See Table 7).

The uptakes from the boiler probably would be 6 in. pipe with a 6 in. × 4 in. reducing ell at top.

[4]*Note.*—As K_1, K_2, K_3 all carry 8000 sq ft and are 6 in. pipe, the whole runout including J_1, J_2 and J_3 and H_1, H_2 and H_3 and the leads from the boiler headers to the main steam header would also be made 6 in. pipe.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

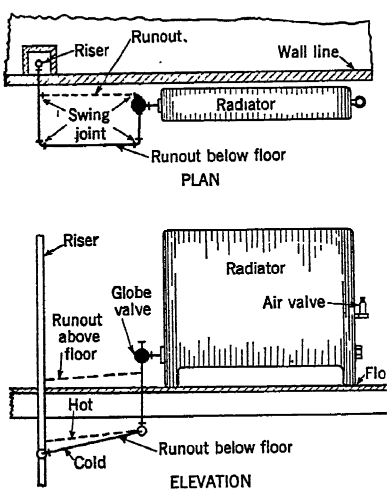


FIG. 7. ONE-PIPE RADIATOR CONNECTIONS

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Pressure Drop for Various Rates of Flow of Water, Fig. 3, Chapter 43. The relative boiler loads should be considered, as in the case of gravity return connections.

Radiator Connections

Radiator connections are important on account of the number of repetitions which occur in every heating installation. They must be properly pitched and they must be arranged to allow not only for movement in the riser but, in frame buildings, for the shrinkage of the building. In a three story building this sometimes amounts to 1 in. or more. The simplest connection is that for the one-pipe system where only one radiator connection is necessary. Where the radiator runouts are located on the ceiling or under the floor, sufficient space usually is available to make

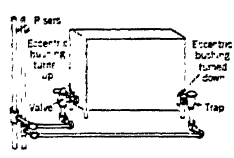


FIG. 8. CONNECTIONS TO STEAM-TYPE RADIATOR FOR TWO-PIPE SYSTEM

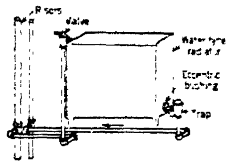


FIG. 9. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

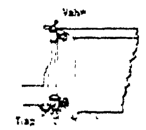


FIG. 10. TOP AND BOTTOM RADIATOR CONNECTIONS

a good swing joint with plenty of pitch, but where the runouts must come above the floor the vertical space is small and the runouts can project out into the room only a short distance. Fig. 7 illustrates two satisfactory methods of making runouts on a one-pipe gravity air vent system of either the up-feed or down-feed type, the runout below the floor being indicated in full lines and the runout above the floor in dotted lines. Sometimes it is necessary to set a radiator on pedestals, or to use high legs, in order to obtain sufficient vertical distance to accommodate above-the-floor runouts. Particular attention must be given to the riser expansion as it will raise the runout and thereby reduce the pitch.

Similar connections for a two-pipe system of the gravity air vent type are illustrated in Fig. 8 for the old steam type radiator. If the water type is used, the supply tapping is at the top instead of at the bottom, the runouts otherwise remaining as shown in Fig. 8. A satisfactory type of radiator connection for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of both the up-feed and down-feed types is shown in Fig. 9.

While short radiators, not exceeding 8 to 10 sections, may be supplied and returned from the same end as indicated in Fig. 10, the top-and-bottom-opposite-end method is to be preferred in all cases where it can be used. On down-feed systems of the atmospheric, vapor, vacuum, sub-atmospheric, and orifice types, the bottom of the supply riser must be dripped into the return somewhat as illustrated in Fig. 11. On up-feed systems of the vapor and atmospheric types, where radiators in the basement are located below the level of the steam main, the drop to the radiator is dripped into the wet return and an air line is used to vent the return radiator connection into an overhead return line, as illustrated in Fig. 12. When the radiator stands on the floor below the main, the drip

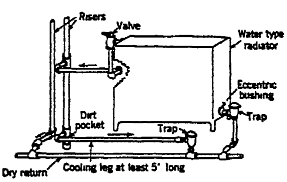


FIG. 11. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

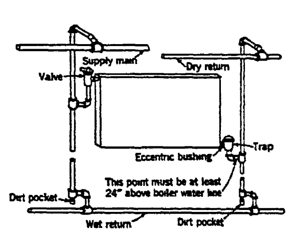


FIG. 12. CONNECTIONS TO RADIATOR HUNG ON WALL

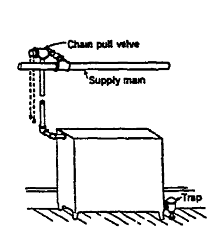


FIG. 13. CONNECTING DROP RISER DIRECT TO RADIATOR

on the steam branch down to the radiator may be omitted if an overhead valve, as shown in Fig. 13, is used. This method is also suitable for vacuum, sub-atmospheric, and orifice systems.

Convector Connections

Convectors often are installed without control valves, a damper being used to shut off the flow of air to retard the heat transfer from the convector even though it is still supplied with steam. The piping connections for a convector with the inlet and outlet at the same end are shown in Fig. 14. There is no valve on the steam side but there is a thermostatic trap on the return. The damper for control is shown immediately above the convector. This piping is suitable for atmospheric, vapor, vacuum,

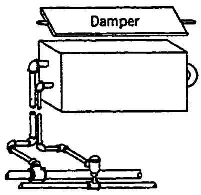


FIG. 14. CONVECTOR CONNECTIONS SAME END

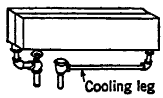


FIG. 15. HORIZONTAL FIN-TYPE HEATING UNIT

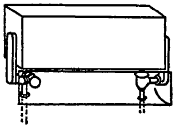


FIG. 16. HEATING UNIT VALVES BEHIND GRILLE

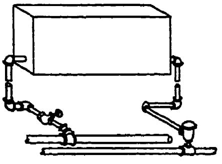


FIG. 17. HEATING UNIT WITH VALVES IN BASEMENT

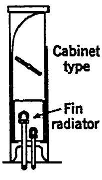


FIG. 18. FIN-TYPE HEATING UNIT IN CABINET

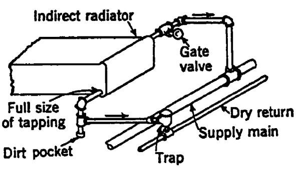


FIG. 19. PIPING CONNECTIONS TO INDIRECT RADIATORS

sub-atmospheric, and orifice systems of the up-feed type. A similar unit with connections on opposite ends and suitable for the same systems is shown in Fig. 15. This unit has no damper but requires a valve on the steam connection for control. When valves must be located so as to be accessible from the supply air grille, the arrangement usually takes the form indicated in Fig. 16. A convector located in the basement and supplying air to a room on the floor above may be piped as pictured in Fig. 17 for all systems except gravity one-pipe or two-pipe systems. Convectors with damper control, installed in cabinets or under window sills, usually are connected as shown in Fig. 18.

Vapor systems with heating units in the basement where the returns are dry would be treated as in Fig. 19. Similar heating units where a wet return is available would be connected as shown in Fig. 20. If the dry

return were on a vacuum, atmospheric, sub-atmospheric or orifice system, the treatment would be identical.

On all heating units it is important to use a nipple the full size of the outlet and to reduce the pipe size to the normal return size required, by the use of a reducing ell, as indicated in Fig. 21.

Pipe Coil Connections

Pipe coils, unless coupled in a correct manner, often give trouble from short circuiting and poor circulation. The method of connecting shown in Fig. 22 is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

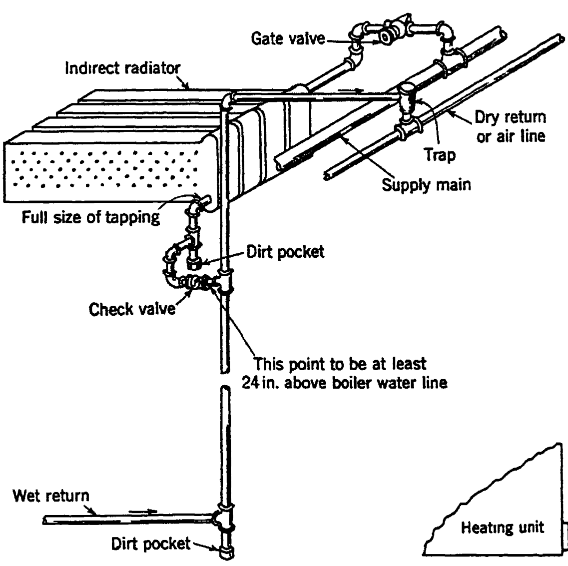


FIG. 20. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

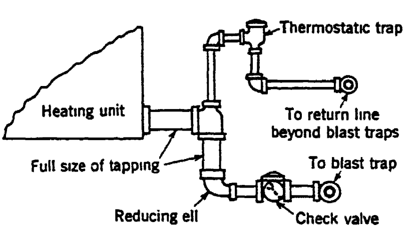


FIG. 21. HEATING UNIT RETURN CONNECTION WITH SEPARATE AIR LINE

Indirect Air Heater Connections

Heating units for central fan systems have simple connections on the steam side. The steam main is carried into the fan room and has a single branch tapped off for each row of heating units. Each of these main branches is split into as many connections as need be made to each row, governed by the number of stacks and the width of the stacks. Each stack must have at least one steam connection, and wide stacks are more evenly heated with two steam connections, one at each end.

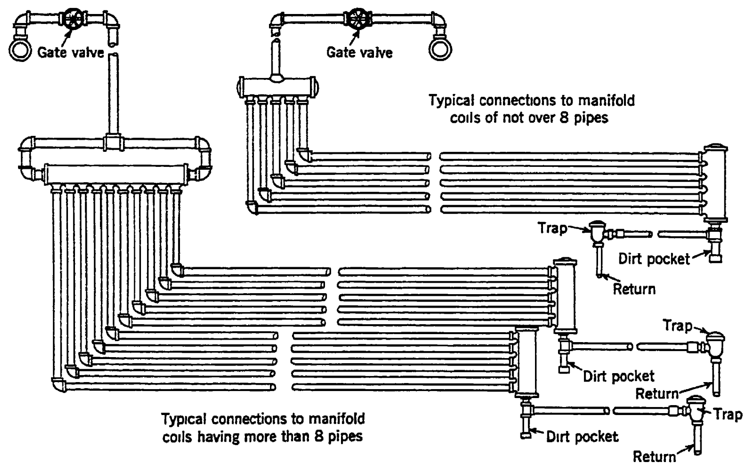


FIG. 22. TYPICAL PIPE COIL CONNECTIONS

The piping shown in Fig. 23 is for small stacks and has the steam connected at only one end. On the return side all of the returns are collected together through check valves and are passed through blast traps which are connected to the vacuum return or to an atmospheric return. The air from the stacks, in the case illustrated, passes up into a small air line and through a thermostatic trap into a line connecting into the return beyond the blast trap.

Where the stacks contain some thirteen or more sections, an auxiliary air tapping is made to the lower portion of one of the middle sections, in the manner illustrated in Fig. 24, to prevent air collecting at this point. Thermostatic control as applied to such heating units in modern practice

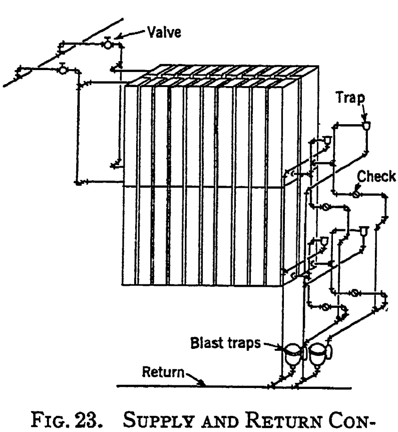


FIG. 23. SUPPLY AND RETURN CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

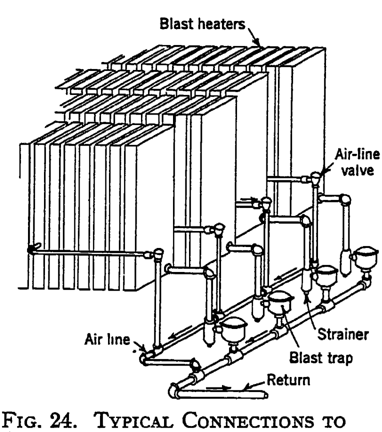


FIG. 24. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

consists of a thermostatic valve located in each main branch from the steam line so that each valve will open or close a complete row of stacks across the entire face of the heating unit. In such cases the outlet connections from each stack should be provided with a check valve. The stack closest to the outside air intake usually is not equipped with a thermostatic valve. A gate valve on the steam pipe to the first coil is operated manually to supply steam continuously in freezing weather. Good practice demands that the returns be connected in parallel with the steam supplies, with a separate steam trap for each bank of coils having a separately valved steam supply. This arrangement is illustrated in Fig. 23, for blast traps having external thermostatic by-passes and integral thermostatic by-passes, respectively.

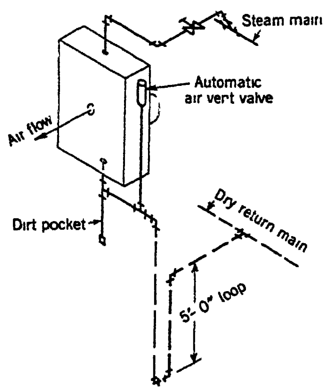


FIG. 25. UNIT HEATER CONNECTED TO ONE-PIPE AIR-VENT SYSTEM

A method of connecting a unit heater to a one-pipe air-vent steam heating system is illustrated in Fig. 25.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \qquad (3)$$

where

EDR = equivalent direct radiation, square feet.

Q = volume of air, cubic feet per minute.

t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.

t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.

60 = the number of minutes in one hour.

55.2 = the number of cubic feet of air heated 1 F by 1 Btu.

240 = the number of Btu in 1 sq ft of EDR.

Example 5. Assume that the heating units shown in Fig. 26 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

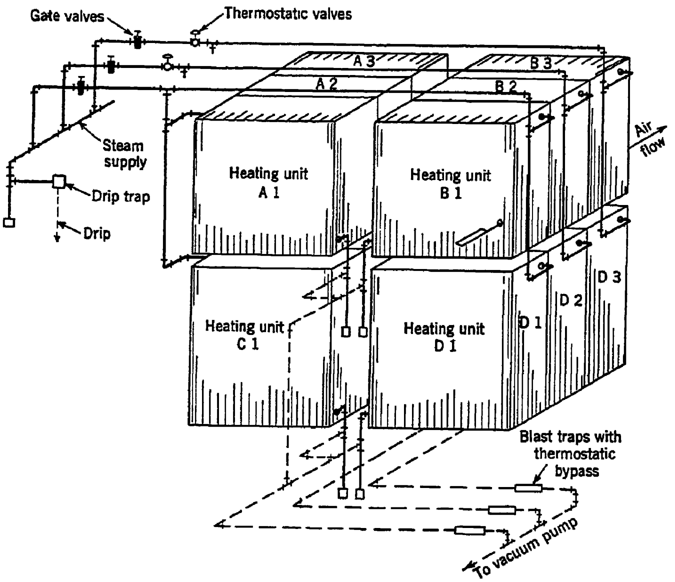


FIG. 26. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	TOTAL LOAD (EDR)	STACK LOAD[a] (EDR)	CONNECTION LOAD[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.

[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

DRIPPING

Any steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the steam flow. Any steam main in any heating system can be elevated if dripped (Fig. 27). Steam mains also may be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for

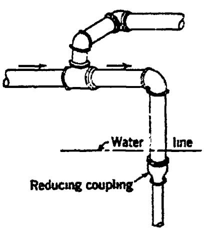


FIG. 27. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

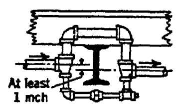


FIG. 28. LOOPING MAIN AROUND BEAM

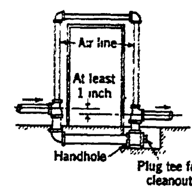


FIG. 29. LOOPING DRY RETURN MAIN AROUND OPENING

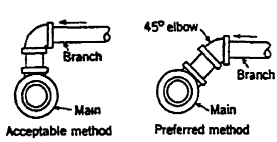
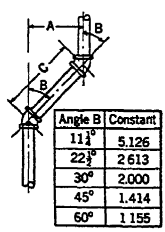


FIG. 30. METHODS OF TAKING BRANCH FROM MAIN



Angle B	Constant
11¼°	5.126
22½°	2 613
30°	2.000
45°	1.414
60°	1 155

To find length C-multiply A by constant for angle B

FIG. 31. CONSTANTS FOR DETERMINING LENGTH OFFSET PIPE

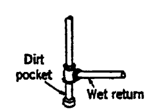


FIG. 32. DIRT POCKET CONNECTION

the condensation (Fig. 28). Return mains may be carried past doorways or other obstructions by using the scheme illustrated in Fig. 29; in vacuum systems it is well to have a gate valve in the air line.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 30, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with

90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 31.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 32.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 33. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems. A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used a cooling leg

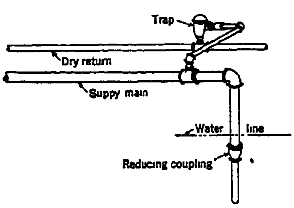


FIG. 33. DRIPPING END OF MAIN INTO WET RETURN

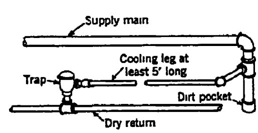


FIG. 34. DRIPPING END OF MAIN INTO DRY RETURN

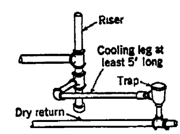


FIG. 35. DRIPPING HEEL OF RISER INTO DRY RETURN

(Fig. 34) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 35. On large systems it is desirable to install a gate valve in the cooling leg ahead of the trap.

PROBLEMS IN PRACTICE

1 ● What factors determine the size of steam piping and the allowable limit of capacity?

Factors which determine the size of steam piping are the desired initial pressure and the allowable drop in pressure which is permissable to maintain a pressure in the farthest radiator. The length of run in sizing piping is important and it is generally considered as the distance along the piping from the source of steam supply to the farthest radiator, with allowances for resistance of elbows and valves expressed in terms of equivalent length.

2 ● When the size of pipe is still undetermined, what arbitrary percentage is usually added to the actual length to obtain the equivalent length?

Usually 100 per cent; in other words, the actual length is doubled to allow for the added drop produced by the valves, tees, elbows, and other fittings.

3 ● What are the major factors to be considered in determining the flow of steam in pipes?

a. The initial steam pressure available and the total pressure drop allowable between the source of steam supply and the end of the return system. The pressure drop should never exceed one half of the initial pressure.

b. The maximum steam velocity allowable. When condensate is flowing against the steam, the velocity must not be so great as to produce water hammer, or hold up water in parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which disturbances take place depends upon:

1. Size of pipe.
2. Whether pipe is vertical or horizontal.
3. Pitch or grade of pipe.
4. Quantity of water flowing against steam.

c. The equivalent length of run from the source of steam supply to the farthest heating unit, with allowance for friction in pipe fittings and valves.

4 ● Name three fundamental considerations in designing the piping system for steam heating.

a. Provision for the distribution of suitable quantities of steam to the various heating units.

b. Provision for the return of condensate from the radiators and piping to the boiler.

c. Provision of means for expelling air from the radiators and piping.

5 ● Why is the proper reaming of the ends of pipe necessary?

The capacities of pipes depend upon the free area available for flow. In cutting the pipe this area may be restricted by a burr, which may decrease the capacity of a pipe more than 25 per cent in the smaller pipe sizes.

6 ● a. What are the major factors to be considered when selecting a pressure reducing valve?

b. How should such valve be installed?

a. The initial pressure of the steam must be considered along with the desired reduced pressure. The connected load to be supplied must be known in square feet of equivalent direct radiation or in pounds of steam per hour. For operation with a continuous load, a semi-balanced or double-seated valve operated by a diaphragm gives good results. Where the load is intermittent, as in process work or with thermostatically controlled blast heaters, a so-called dead end or single-seated valve should be used.

b. The pressure reducing valve should be installed in a horizontal line with a gate valve on each side, and with a by-pass operated by a valve. The pressure balancing pipe from the diaphragm chamber should be connected into the top or side of the low pressure main not less than 15 ft from the reducing valve.

7 ● What is the usual expansion allowance and how it is compensated for in heating system supply risers?

The expansion of low pressure steam piping is normally taken as 1¼ to 1½ in. per 100 ft of pipe. With a five story building a double swing connection between the riser and the main will suffice. In buildings between 5 and 10 stories high the riser should be anchored near its center and have double swing connections to the main. For taller buildings expansion loops or riser offsets are used which are capable of handling a length of riser reaching 5 stories in either direction from the joint. The risers are anchored at each alternate 5 stories. All radiators must have double swing connections, and those connected above where the riser is anchored must be given greater pitch to insure their having proper grade when the riser is heated.

8 ● Why should all boiler steam supply tappings be used full size?

In order to operate at low steam velocities so the water in suspension can separate from the steam and remain in the boiler.

9 ● What is the Underwriters Loop or the Hartford Connection?

An arrangement of piping on the returns to low pressure boilers wherein the return line is raised up nearly to the water line of the boiler and is then dropped back and connected to the boiler return inlet; the high point is connected by a balanced pipe to the steam runout from the boiler on the boiler side of all stop valves. With this loop no check valve is required on gravity systems, and water cannot be backed out of the boiler and into the return at a point lower than the invert of the pipe at the top of the loop.

10 ● What are the important factors in making radiator connections?

Connections to radiators should be made as direct as possible, of proper size, with ample pitch of piping and allowance for expansion.

11 ● Why should careful attention be given to proper dripping and drainage of steam piping?

The steam mains and risers must be quickly drained of condensate and where necessary vented of air in order to obtain a sufficient supply of steam to the radiators. Proper drainage is also necessary to insure a noiseless heating system.

12 ● What is the limit of pressure drop usually recommended in a vacuum system?

Not over $\frac{1}{8}$ lb (2 oz) per 100 ft of equivalent run, and not over 1 lb total drop.

13 ● When steam and condensation are flowing in the same direction, what is the maximum total pressure drop which should be used?

The maximum total pressure drop should not exceed one half of the initial steam pressure.

14 ● What does a proper installation of a pressure reducing valve include?

A strainer in front of the pressure reducing valve; a gate valve in front of the strainer; a gate valve after the reducing valve; a by-pass around the two gate valves, strainer, and pressure reducing valve; and a globe valve in the by-pass. Sometimes a safety valve on the low pressure side and pressure gages on both sides are installed. The high pressure line should be dripped just before the high pressure steam enters the pressure reducing valve assembly.

15 ● Will a pressure reducing valve which is reducing the steam pressure from 100 lb gage to 50 lb gage pass more or less steam than the same valve when reducing the steam pressure from 100 lb gage to 5 lb gage?

The valve will pass practically the same volume of steam in each case as the velocity of steam flowing through an orifice shows no material increase after the reduced absolute pressure has fallen to 58 per cent of the initial absolute pressure. Because of its greater density, the weight of steam passed will be greater in the case of the reduction to 50 lb gage.

Chapter 17

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Mechanical Circulation, Circulators, Iron Pipe and Copper Tube Sizes, Gravity Circulation, Expansion Tanks, Relief Valves, Installation Details

THE various forms of hot water heating may be fundamentally classified according to motive force, namely, forced circulation or gravity flow. Forced circulation is accomplished by the use of centrifugal or propeller type pumps which are especially designed for this particular type of application. Gravity flow is maintained by the difference in weight of the water in the flow and return mains.

These systems may be further classified as to high or low operating water temperatures. Higher water temperatures permit a reduction in radiator size. A large temperature differential between the flow and return results in smaller pipe sizes as also does the use of forced circulation. Light wall copper tubing has recently been introduced to supplement the customary black iron piping which has been used for these systems in the past.

Low temperature water(150 to 180 F) is generally that which provides a heat emission per square foot of radiation of from 150 to 165 Btu while a high temperature water (200 to 220 F) will deliver from 200 to 240 Btu.

The use of high temperature water in a heating system is desirable as the maximum outside temperature for which the system is designed will occur for a relatively short time during the average season. The increased use of automatic heating equipment with more accurate controls, makes it possible to use higher temperatures and smaller heating units without sacrificing good design.

The unit, a square foot of equivalent direct radiation, EDR, has been used for many years for rating purposes in both steam and hot water systems, but its use, especially in hot water systems, has always resulted in complications and confusion. It is the plan of THE GUIDE *to eventually eliminate this empirical expression and to substitute a logical unit based on the Btu. The* Mb, *the equivalent of 1000 Btu and the* Mbh, *the equivalent of 1000 Btu per hour, which have been approved by the* A.S.H.V.E. *are used in this chapter on hot water systems to replace the square foot of radiation formerly used.*

In designing a piping arrangement for a hot water heating system, it is necessary to observe the fundamental rule that the total friction head in any circuit must not exceed the pressure head available for circulating the water. It is necessary to size the pipe in any circuit, so that the friction loss produced by the movement of a sufficient volume of water to handle the heating load will not be greater than the available head.

In designing a hot water heating system, it is necessary to determine:

1. The heat losses of the rooms or spaces to be heated. (See Chapter 7.)
2. The size and type of boiler. (See Chapter 13.)
3. The location, type, and size of heating units. (See Chapter 14.)
4. The method of piping.
5. The type and size of circulating pump (if forced circulation).
6. Suitable pipe sizes.
7. The type and size of expansion tank.

ONE- AND TWO-PIPE SYSTEMS

Piping systems may be divided into two general types, namely, one-pipe and two-pipe systems. These fundamental piping layouts may differentiate between up-flow, down-flow and zoned systems. Also the type of riser and radiator connection may vary considerably. Zoning is important in modern design and it is accomplished by dividing the system into a number of circuits and controlling each circuit individually. In a two-pipe system the piping is arranged so that the water flows through only one radiator during a circuit through the system, so that all radiators are supplied with water at practically the same temperature as that in the boiler. In some one-pipe systems, the water flows through more than one radiator during its circuit. In that case, the first radiator receives the hottest water; the second radiator, somewhat cooler water; the third one, still cooler; and so on. As the temperature of the water supplied to a radiator is lowered, the size of the radiator must be increased and, consequently, the total heating surface for a one-pipe system must be greater than for a two-pipe system for the same requirements. As the velocity is *increased* in a one-pipe system, the drop in temperature is *decreased*, so that water at a higher average temperature is delivered to the radiators. This means that the radiators at the end of the main can be sized on the same basis as the radiators at the beginning of the main. If the system is correctly designed, the resulting error is less than the variation in calculating the heating load for the enclosure.

By making use of improved devices now available, one-pipe forced circulation systems may be calculated by the same procedure described later for two-pipe systems. Operation may be obtained as satisfactory as with a two-pipe system.

Two-pipe systems may be divided into two classes, *direct return* systems (Fig. 1), and *reversed return* systems (Fig. 2). In a direct return system the water returns to the heater by a direct route after it has passed through its radiator and, as a result, the paths through the three radiators shown in Fig. 1 are of unequal lengths, the path through the first radiator being the shortest and that through the third radiator, the

longest. In a reversed return system, the water returns to the heater by an indirect route after it has passed through the radiators, so that the paths leading through the three radiators shown in Fig. 2 are practically of equal length.

The reversed return system has an advantage over the direct return system in that it is more likely to function satisfactorily even though the pipe system is not accurately designed. For example, if in Fig. 2 all pipes are of one size, each of the three radiators will receive approximately the same quantity of hot water because the three paths are practically of equal length, whereas in Fig. 1, if all pipes are of the same size, Radiator 1 will receive more water than the others because the path through it is shorter than those through the other radiators. As a result, Radiator 1 will be filled with water at a higher average temperature than the remaining two radiators, and will therefore dissipate more heat. To prevent this unequal distribution of heat it is necessary to throttle the paths through Radiators 1 and 2 so that the friction heads of the three paths are equal when each radiator receives its proper quantity of water.

The two-pipe direct return system, with its inherent lack of balance, is the least satisfactory type of piping possible, yet is the most widely used.

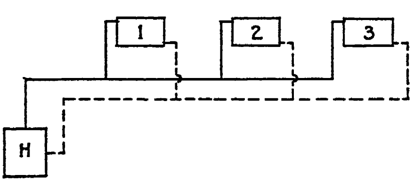


FIG. 1. A DIRECT RETURN SYSTEM

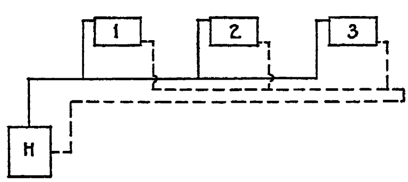


FIG. 2. A REVERSED RETURN SYSTEM

The modern applications of automatic heating require a system to be very nearly in balance so that uniform distribution of heat will be obtained.

Two-pipe systems must be balanced first by calculation and then by test after the plant is in operation. Unbalanced conditions in a forced circulation system are more detrimental to satisfactory operation than in the system circulated by gravity. The selection of orifices for correcting the unbalance must be more accurate. Due to the variations in water delivery from pipes, the accuracy of calculations is decreased, so that more reliance must be placed on actual test work. This is always costly and seldom completely satisfactory.

A comparison of Fig. 1 and Fig. 2 may suggest that a reversed return system requires considerably longer mains than a direct return system. This is not always the case, as will be noted from the reversed return system of Fig. 3.

MECHANICAL CIRCULATION AND CIRCULATORS

The designer of a forced circulation system generally makes use of the pumps commercially available. Pumps of this type will have characteristics which govern the water velocity selected for the heating system. However, available pumps generally have a sufficient range of capacities

to promote the selection of an economical velocity. If a system is designed to handle a load of 96 Mbh with a 20 F drop allowable in the system, a circulating pump will be required, handling about 10 gpm and at a head pressure high enough to allow a satisfactory friction drop in the system.

Frequently water velocities are selected which produce objectionable noises in the system. A velocity of over 4 fps is apt to cause noise in the smaller pipes and tubes. Velocities higher than this value will cause no objectionable trouble in industrial applications.

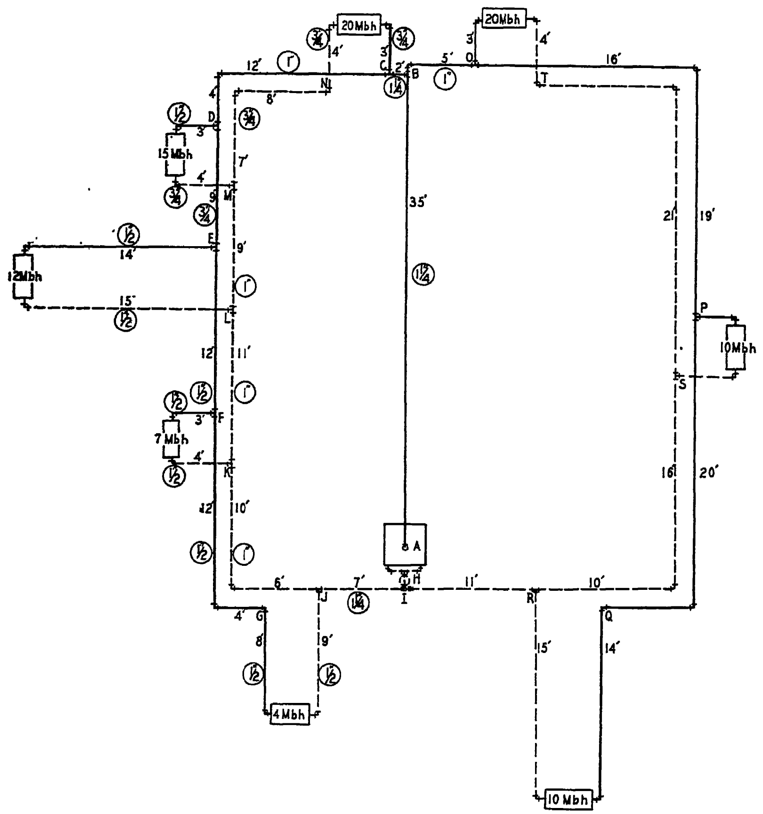


FIG. 3. A FORCED CIRCULATION REVERSED RETURN SYSTEM[a]

[a]Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet.

Low head centrifugal pumps especially designed for hot water systems are used to provide the necessary head pressure for forced circulation and to improve the operation of an improperly designed or installed gravity system. These pumps are designated by the nominal pipe size of their connection, but the selection of the pump should be governed by the capacity curves and not by the nominal pipe size. These pumps operate with little noise and low power consumption, both of which are features of prime importance to the satisfactory operation of a forced

circulation system. They are designed for installation directly into the heating main and require no other support. The common practice is to install them in the return line but where desirable there is no objection to their location in the supply line. Gate valves should be installed in either side of the pump so that it can be removed without draining a system. A by-pass is not necessary as the friction drop through the pump is not sufficient to prevent gravity recirculation if the pump should become inoperative.

Propeller type pumps are also available for hot water service, generally being built into a fitting and are made in all of the commercial pipe sizes commonly used in heating. They are installed in the same manner as a centrifugal pump.

Forced circulation lends itself to automatic control and the arrangement of the circuit depends entirely on the design of the system. The control may consist of a thermostat controlling both the automatic firing device and the circulator with the same type of limit control, as a safety switch. This type of control can be satisfactory, provided the radiation is properly selected and accurately located in the building. A circuit using flow control valves to regulate the gravity flow of the water when the pump is not running allows the temperature to be maintained closer to the desired setting. Under these circumstances, the circulator motor is controlled by a room thermostat while the automatic firing device is controlled by a limit switch with a safety device in series.

For exceptionally large installations, such as central heating plants circulating pumps of the centrifugal single stage type having an average operating efficiency of 70 per cent against heads up to 125 ft are sometimes used. In some cases it is advisable to install pumps in duplicate to provide for contingencies and to insure continuous operation. In such cases, each pump should be made equal to the maximum capacity required.

PIPE SIZES

The pressure heads available in forced circulation systems are much greater than those in gravity circulation systems, consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipes of a heating system are reduced in size, the necessary increase in the velocity of the water increases both the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible control of hot water circulation. This improved performance merits the small increase in operating cost necessary to mechanically circulate the system. The velocity required should be determined by calculation for the particular system under consideration.

Since the velocities in forced circulation systems are higher than those in gravity circulation systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system and, consequently, it

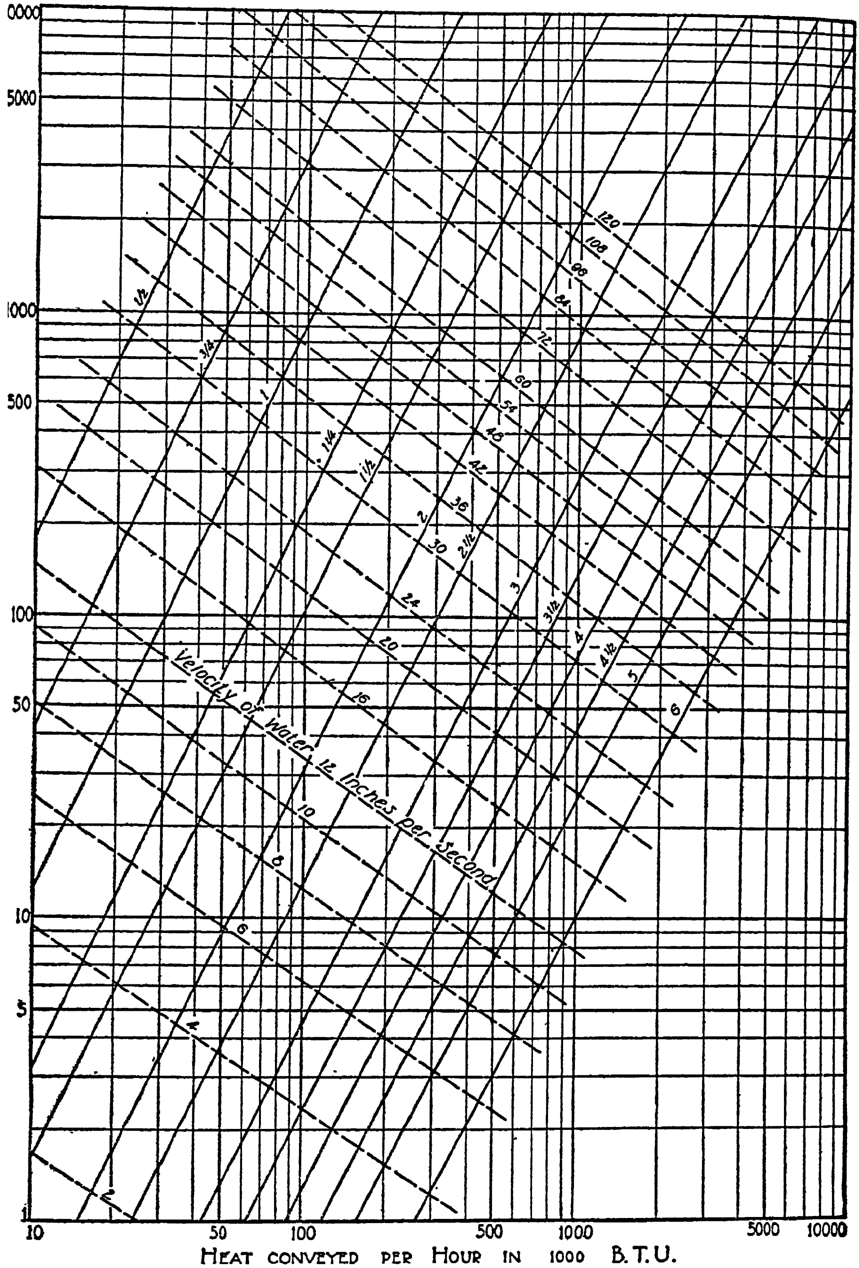


FIG. 4. FRICTION HEADS IN BLACK IRON PIPES FOR A 20 F TEMPERATURE DIFFERENCE OF THE WATER IN THE FLOW AND RETURN LINES

is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FORCED CIRCULATION

In designing a forced circulation system, black iron pipe sizes may be selected from either Fig. 4 or Table 1, both of which are based on a 20 F temperature difference between the flow and return lines. For other temperature drops, the pipe capacities may be changed to correspond to the desired differentials. Research data are lacking for determining the capacities of copper tube sizes. In the absence of complete test data at the present time, capacities are given in Table 2 for type L copper tube sizes which are based on a recently developed hydraulic formula[1]. The friction heads of boiler, radiator valve and tee may be expressed in terms of friction head in one elbow according to the values given in Table 3 for iron pipe, and Table 4 for copper tubing.

The following examples will illustrate the procedure to be followed in designing forced circulation systems.

Example 1. From the plan of Fig. 3 note that the longest circuit consists of 151 ft of iron pipe; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; 10 ells and 3 tees; and the shortest circuit consists of 127 ft of pipe; 4 tees; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; and 6 ells. Design the piping for this system.

Solution. The friction in the various fittings can be expressed in terms of the friction in a 90-deg elbow from the values given in Table 3. The longest circuit consists of 151 ft of pipe and 44 elbow equivalents. The short circuit consists of 127 ft of pipe and 39 elbow equivalents.

The friction head in one elbow is approximately equal to the friction produced by the same sized pipe 25 diameters in length. Assume that the average pipe size for this system is 1 in. The equivalent length of the longest circuit will be 151 ft plus 100 ft or 251 ft of pipe. The equivalent length of the short circuit will be 217 ft.

Having determined the equivalent length of the circuits, the next step is to assume the rate at which the water is to be circulated in the system. The water may flow through the system so that it will cool any reasonable number of degrees. For the most economical average system a 20 F drop seems to be a satisfactory rate. This entails a slower water flow from the pumping equipment with a reasonable relationship between pipe size and flow. Assume 20 F drop for this system. One gallon of water per minute with a density of 7.99 at 215 F will deliver approximately 9600 Btu per hour with a 20 F drop. The total radiation load is 98 Mbh, therefore the pump must deliver 10.2 gpm or 4900 lb of water per hour.

Knowing that the rate of flow is 10.2 gpm, the next step is to determine from the characteristics of available pumps, which one will produce a satisfactory velocity in the system. Assume that 4 pumps are available for this load which will produce 10.2 gpm at pressure heads of 2, 5, 10 and 18 ft. At these heads the pumps would produce a velocity high enough to make available a friction head per foot of pipe of 96, 240, 480 and 860 milinches per foot respectively. If 95 milinches per foot were used, the gravity head at 215 F average temperature in the mains would be 26 per cent of the total head and should be considered in sizing the system. At 240 milinches per foot the gravity effect is 10 per cent and as this is lower than the delivery variation from the pipe used, it can be neglected. At 480 and 860 milinches the gravity effect is still a smaller percentage of the total, but at these losses in the average system the cost of pumping will more than offset the advantage gained in pipe sizes. Therefore, pipe size this system at 240 milinches per foot which is equivalent to a total loss of 60,000 milinches for the 250 ft equivalent length of pipe.

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[1]Hydraulic Service Characteristics of Small Metallic Pipes, by G. M. Fair, M. C. Whipple and C. Y. Hsiao (*Journal of the New England Water Works Association*, Vol. XLIV, No. 4, 1930).

TABLE 1. CAPACITIES FOR BLACK IRON PIPE

A = *Carrying capacities in Mbh*
B = *Velocity in inches per second*

Head Loss, Ft	Milinch Friction Loss Per Foot of Pipe: 720	480	360	300	240	180	160	144	120	96	90	80	70	60
	Equivalent Length of Pipe in Feet (Longest Circuit)													
2	33	50	66	80	100	133	150	167	200	250	270	300	340	400
2½	42	62	84	100	125	167	188	208	250	312	333	375	428	500
3	50	75	100	120	150	200	225	250	300	375	400	450	510	600
3½	59	87	117	140	175	233	263	291	350	437	463	525	593	700
4	67	100	133	160	200	266	300	333	400	500	533	600	685	800
4½	75	112	149	180	225	300	338	374	450	562	593	675	758	900
5	83	125	167	200	250	333	375	416	500	625	666	750	860	1000
5½	92	137	183	220	275	366	413	457	550	687	713	825	923	1100
6	100	150	200	240	300	400	450	500	600	750	800	900	1030	1200
6½	108	162	217	260	325	433	488	540	650	812	843	975	1088	1300
7	116	175	233	280	350	465	525	580	700	875	933	1050	1200	1400
7½	124	187	249	300	375	500	563	623	750	937	973	1125	1252	1500
8	133	200	266	320	400	533	600	666	800	1000	1070	1200	1370	1600
8½	142	212	283	340	425	566	638	706	850	1062	1103	1275	1417	1700
9	150	225	300	360	450	600	675	750	900	1125	1200	1350	1540	1800
9½	159	237	317	380	475	633	713	789	950	1187	1233	1425	1577	1900
10	167	250	333	400	500	666	750	833	1000	1250	1333	1500	1715	2000
10½	175	262	349	420	525	700	788	872	1050	1312	1363	1575	1737	2100
11	183	275	366	440	550	733	825	916	1100	1375	1466	1650	1885	2200
11½	192	287	383	460	575	766	863	955	1150	1437	1533	1725	1897	2300
12	200	300	400	480	600	800	900	1000	1200	1500	1600	1800	2030	2400
Nominal Pipe Size, In.	Capacity of Pipes *Mbh* With a 20 F[a] Drop													
½ A	20	16	14	13	11	10	9	9	8	7	7	6	6	5
½ B	*27*	*22*	*19*	*17*	*15*	*13*	*12*	*11*	*10*	*9*	*9*	*8*	*8*	*7*
¾ A	43	35	30	27	24	21	19	18	17	15	14	13	12	11
¾ B	*33*	*25*	*23*	*21*	*18*	*16*	*15*	*14*	*13*	*11*	*11*	*10*	*9*	*9*
1 A	85	70	60	54	48	41	39	36	33	30	28	27	25	23
1 B	*39*	*32*	*27*	*25*	*22*	*19*	*18*	*17*	*15*	*13*	*13*	*12*	*11*	*10*
1¼ A	180	145	125	115	98	85	80	75	68	60	58	55	51	47
1¼ B	*48*	*39*	*33*	*30*	*27*	*23*	*21*	*20*	*19*	*16*	*15*	*15*	*14*	*12*
1½ A	285	230	195	180	160	135	125	120	110	96	92	88	82	75
1½ B	*54*	*44*	*38*	*34*	*30*	*26*	*24*	*23*	*21*	*19*	*18*	*17*	*15*	*14*
2 A	540	435	370	340	300	255	240	230	205	180	175	165	150	140
2 B	*64*	*52*	*45*	*40*	*36*	*30*	*29*	*27*	*24*	*22*	*21*	*20*	*19*	*17*
2½ A	890	720	610	550	480	420	390	370	330	300	280	270	250	230
2½ B	*74*	*60*	*50*	*46*	*41*	*35*	*33*	*31*	*28*	*24*	*24*	*22*	*21*	*19*
3 A	1650	1340	1130	1000	900	760	720	670	600	540	520	480	450	410
3 B	*88*	*70*	*60*	*54*	*48*	*41*	*38*	*36*	*33*	*29*	*28*	*26*	*24*	*22*
3½ A	2500	2000	1700	1500	1350	1150	1080	1000	900	800	760	720	670	620
3½ B	*99*	*78*	*66*	*60*	*54*	*46*	*43*	*40*	*36*	*32*	*31*	*29*	*27*	*25*
4 A	3500	2800	2400	2200	1900	1600	1520	1440	1300	1150	1100	1050	960	880
4 B	*110*	*87*	*74*	*66*	*58*	*50*	*47*	*45*	*40*	*35*	*34*	*32*	*30*	*27*
5 A	7000	5600	4700	4300	3700	3200	3000	2750	2500	2200	2100	2000	1800	1700
5 B	*132*	*106*	*90*	*80*	*70*	*60*	*56*	*53*	*48*	*42*	*41*	*38*	*35*	*32*
6 A	12,000	9200	7800	7000	6200	5200	4800	4600	4100	3600	3500	3300	3000	2800
6 B	*156*	*124*	*104*	*94*	*82*	*69*	*64*	*61*	*55*	*48*	*46*	*44*	*41*	*37*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

TABLE 2. CAPACITIES FOR TYPE *L* COPPER TUBE

A = *Carrying capacity in Mbh*

B = *Velocity in inches per second*

HEAD LOSS FT	MILINCH FRICTION LOSS PER FOOT OF TUBE											
	720	600	480	360	300	240	180	150	120	90	75	60
	EQUIVALENT LENGTH OF TUBE IN FEET (LONGEST CIRCUIT)											
2	33	40	50	67	80	100	133	160	200	267	320	400
2½	42	50	63	83	100	125	167	200	250	333	400	500
3	50	60	75	100	120	150	200	240	300	400	480	600
3½	58	70	88	117	140	175	233	280	350	467	560	700
4	67	80	100	133	160	200	267	320	400	533	640	800
4½	75	90	113	150	180	225	300	360	450	600	720	900
5	83	100	125	167	200	250	333	400	500	667	800	1000
5½	92	110	138	183	220	275	367	440	550	733	880	1100
6	100	120	150	200	240	300	400	480	600	800	960	1200
6½	108	130	163	217	260	325	433	520	650	867	1040	1300
7	117	140	175	233	280	350	467	560	700	933	1120	1400
7½	125	150	188	250	300	375	500	600	750	1000	1200	1500
8	133	160	200	267	320	400	533	640	800	1067	1280	1600
8½	142	170	213	283	340	425	567	680	850	1133	1360	1700
9	150	180	225	300	360	450	600	720	900	1200	1440	1800
9½	159	190	238	317	380	475	633	760	950	1267	1520	1900
10	167	200	250	333	400	500	667	800	1000	1333	1600	2000
10½	175	210	263	350	420	525	700	840	1050	1400	1680	2100
11	183	220	275	367	440	550	733	880	1100	1467	1760	2200
11½	192	230	288	383	460	575	767	920	1150	1533	1840	2300
12	200	240	300	400	480	600	800	960	1200	1600	1920	2400
NOMINAL TUBE SIZE, IN.	CAPACITY OF TUBES *Mbh* WITH A 20 F[a] DROP											
⅜ A	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
⅜ B	*27*	*24*	*21*	*18*	*16.5*	14	*13*	11	*10*	*8.5*	*8*	*7*
½ A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
½ B	*33*	*30*	*25*	*21*	*19*	*17*	*15*	*13*	*12*	*10*	*9*	*8*
⅝ A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
⅝ B	*37*	*34*	*30*	*24*	*21*	*19*	*17*	*15*	*13*	*11*	*10*	*9*
¾ A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
¾ B	*42*	*38*	*33*	*27*	*24*	*21*	*19*	*17*	*14*	*12*	*11.5*	*10*
1 A	104	94	82	70	63	56	47	42	37	32	28	25
1 B	*48*	*45*	*39*	*34*	*30*	*25*	*22*	*19*	*17*	*14.5*	*13*	*12*
1¼ A	185	169	149	125	112	100	84	75	66	56	50	44
1¼ B	*55*	*51*	*45*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*	*13*
1½ A	300	270	235	200	180	160	134	120	105	90	81	71
1½ B	*62*	*57*	*51*	*43*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*
2 A	625	560	495	420	375	335	280	250	200	188	170	150
2 B	*76*	*68*	*59*	*51*	*47*	*42*	*36*	*32*	*27*	*22*	*20*	*18*
2½ A	1130	1010	890	750	680	600	500	450	395	335	305	270
2½ B	*90*	*80*	*69*	*58*	*49*	*47*	*42*	*37*	*33*	*26*	*23*	*21*
3 A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
3 B	*98*	*90*	*80*	*66*	*59*	*52*	*47*	*42*	*36*	*30*	*27*	*23*
3½ A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
3½ B	*110*	*100*	*89*	*75*	*66*	*57*	*51*	*45*	*40*	*35*	*30*	*26*
4 A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
4 B	*120*	*108*	*96*	*83*	*73*	*63*	*55*	*49*	*44*	*37*	*34*	*29*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

TABLE 3. IRON ELBOW EQUIVALENTS[a]

1 90-deg elbow	1.0
1 45-deg elbow	0.7
1 90-deg long turn elbow	0.5
1 open return bend	1.0
1 open gate valve	0.5
1 open globe valve	12.0
1 angle radiator valve	2.0
1 radiator	3.0
1 boiler or heater	3.0
1 tee	(Note[b])

[a]The loss of head in one elbow can be expressed in terms of the velocity head by the formula:

$$h = \frac{v^2}{2g} \quad (1)$$

where

h = the loss of head in feet, v = the velocity of approach in feet per second, and $2g$ = 64.4 ft per second per second.

[b]The loss of head in tees when water is diverted at right angles through a branch of the tee varies with the per cent diverted. When the water diverted is less than 60 per cent of that approaching the tee, the loss of head, in elbow equivalents, may be expressed as follows:

$$h_e = \frac{v_1^2}{v_2^2} \quad (2)$$

where

h_e = the loss of head in elbow equivalents, v_1 = the velocity of approach, v_2 = the velocity of water diverted at right angles.

Values in elbow equivalents for the most common percentages of water diverted in a 1x1x1-in. tee are as follows:

25 per cent	16.0
33 per cent	9.0
50 per cent	4.0
100 per cent	1.8

TABLE 4. COPPER ELBOW EQUIVALENTS[a]

90-deg elbow	1.0
45-deg elbow	0.7
90-deg long turn elbow	0.5
open return bend	1.0
open gate valve	0.7
open globe valve	17.0
radiator valve	3.0
radiator	4.0
boiler or heater	4.0
tee	(Note[b])

[a]The loss of head in an elbow can be expressed in terms of the velocity head by the formula:

$$h = \frac{0.7\, v^2}{2g} \quad (3)$$

where

h = loss of head in milinches, v = velocity in inches per second, and g = acceleration of gravity (386 in. per second per second).

[b]The loss of head in copper tees:

$$N = 0.7\,\frac{(v_1^2 + v_3^2)}{v_3^2} \quad (4)$$

where

N = number of elbows that would cause the same loss as the tee when the velocity of water in the connecting pipe is v_3,
v_1 = velocity of the water in the pipe entering the tee, and
v_3 = velocity of the water in the pipe discharging from the tee at right angles to v_1.

Values in elbow equivalents for most common percentages of water diverted in a 1 in. x 1 in. tee.

100 per cent	1.2
50 per cent	4.0
30 per cent	16.0
25 per cent	20.0

The pipe sizes may be selected from Fig. 4 or from Table 1 which has been derived from Fig. 4.

Size the supply main of the longest circuit first. Section AB carries 98 Mbh. From Fig. 4 it will be noted that at 240 milinches per foot, a 1¼ in. pipe carries 98 Mbh. Therefore, use 1¼ in. pipe in Section AB. Section BO carries 40 Mbh. A 1 in. pipe carries 48 Mbh at 240 milinches per foot. Use a 1 in. pipe. Section OP carries 20 Mbh and this will require ¾ in. pipe. Section PQ carries 10 Mbh and requires ½ in. pipe. To size the return start from the boiler and proceed backwards. Section IR carries 40 Mbh and from Fig. 4 a 1 in. pipe is required. Section RS carries 30 Mbh which is only slightly over the capacity of a ¾ in. pipe, so use ¾ in. Section ST carries 20 Mbh and requires a ¾ in. pipe. The radiator branches are determined in the same manner. It is evident from the chart that it is impossible to maintain a constant friction loss per foot and therefore as the delivery varies there will be a change in the desired friction loss per foot of pipe.

It is desirable to check the various circuits so that if the variation from the calculated resistance is too great, it may be compensated by adding additional resistance at the proper point. This may be accomplished by sizing the short circuits by the procedure previously outlined. Prepare a chart such as Table 5 to be used in calculating the resistance of each circuit.

Section AB carries 98 Mbh with a unit head of 240 milinches per foot. In section AB there are 37 ft of pipe and 1¼ in. elbow. At 240 milinches per foot this is equivalent to 9600 milinches total loss in this section. Section BC carries 58 Mbh with a length of 2 ft and 4 elbows. The unit loss in this section is 90 milinches per foot. Loss in this section is then 1080 milinches. Section CD carries 38 Mbh and has 16 ft of pipe and 1 elbow. The unit loss in 1 in. pipe is 155 milinches. The loss in this section is 2790 milinches. The balance of the supply main and the return main are handled in a similar manner.

TABLE 5. PIPING CHECK CHART

LOAD, *Mbh*			PIPE LENGTH FT	ELBOWS	PIPE SIZE IN.	UNIT HEAD MILINCHES PER FT	FRICTION MILINCHES	TOTAL LOSS MILINCHES
Supply Main								
AB	98		37	1	1¼	240	9600	9,600
BC	58		2	4	1¼	90	1080	10,680
CD	38		16	1	1	155	2790	13,470
DE	23		9	0	¾	220	1980	15,450
EF	11		12	0	½	240	2880	18,330
FG	4		16	1	½	50	850	19,180
Return Main								
HI	98		5	5	1¼	240	4320	4,320
IJ	58		11	1	1¼	90	1260	5,580
JK	54		16	1	1	300	5400	10,880
KL	47		11	0	1	230	2530	13,410
LM	35		9	0	1	140	1260	14,670
MN	20		15	1	¾	170	2890	17,560
Radiator Circuits								
CN	20	*Supply*	3	13	¾	170	3910	
		Return	4	2	¾	170	1190	5,100
DM	15	*Supply*	3	19	½	420	9250	
		Return	4	17	¾	96	2880	12,130
EL	12	*Supply*	14	20	½	270	9180	
		Return	15	20	½	270	9450	18,630
FK	7	*Supply*	3	19	½	100	2200	
		Return	4	17	½	100	2100	4,300
GJ	4	*Supply*	8	5	½	50	650	
		Return	9	17	½	50	1300	1,950

The radiator circuits are then checked. The 20 Mbh radiator on this circuit has 3 ft of supply pipe and 13 elbow equivalents while the return is composed of 4 ft and 2 elbows. The unit loss in ¾ in. pipe at this delivery is 170 milinches per foot. The total loss in the supply is 3910 milinches. The loss in the return is 1190. Total loss in the radiator circuit is 5100 milinches. Check each radiator circuit in a similar manner.

The total calculated loss for the longest circuit was determined as 60,000 milinches. The maximum loss in the short circuit is 18,630 plus 13,410 plus 15,450 or a total of 47,490 milinches. This difference is caused by the variation in length of the two circuits and may be corrected by using a flow control in the return main to supply the additional resistance or by introducing resistance into each separate circuit to compensate for the difference. A 10 per cent variation will cause no complication as the flow from the various pipes will not exactly follow the curves of Fig. 4 any closer than this value.

Example 2. Design a two-pipe direct return forced circulation system with copper tubing and fittings for the piping layout as detailed in Fig. 5, based on a 20 F temperature drop through the radiation.

The piping circuit from the boiler to the highest radiator on the farthest riser and back to the boiler is 250 ft of pipe. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so that the total equivalent pipe length is 300 ft.

Assume that a circulator is available which will provide a pressure head of 6 ft.

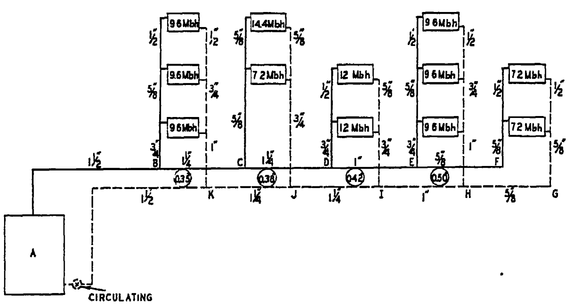


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

Solution. Refer to Table 2, which indicates the total equivalent lengths for pressure heads from 2 to 12 ft. With a circulator having a 6 ft pressure head and a system with a total equivalent length of 300 ft, the piping system will be designed on a basis of 240 milinch.

Checking the piping diagram it will be noted that sections AB and KA, both supply 117.6 Mbh. Referring to the 240 milinch column of Table 2, 1½ in. is shown to be the necessary pipe size. Sections BC and JK carry 88.8 Mbh and require 1¼ in. tubing. Sections CD and IJ supply 67.2 Mbh and require 1¼ in. tubing. Sections DE and HI supply 43.2 Mbh, which requires 1 in. tubing. Sections EF and GH with a load of 14.4 Mbh require ⅝ in. tubing.

The risers are pipe sized in a similar manner. To secure proper distribution of hot water in the direct return system among the several risers, it is necessary to introduce resistances to balance the circuit.

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, that is, operated under equal pressure heads, resistance must be added to the first riser equal to the friction head in the 80 ft of supply main B to F plus the 80 ft of return main G to K for a total of 160 ft of pipe.

Having designed the piping system on a 240 milinch basis, the total friction head in the supply and return mains between the first and fifth risers is therefore 160 × 240 = 38,400 milinches, or 3.2 ft which must be supplied by additional resistance in the first riser.

TABLE 6. FRICTION HEADS (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

DIAMETER OF ORIFICES (INCHES)	VELOCITY OF WATER IN PIPE IN INCHES PER SECOND									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois.(*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

This resistance can be supplied by a calibrated and adjusted modulating valve or by an orifice resistor in a union. If the orifice resistor is to be used, its size may be selected from Table 6.

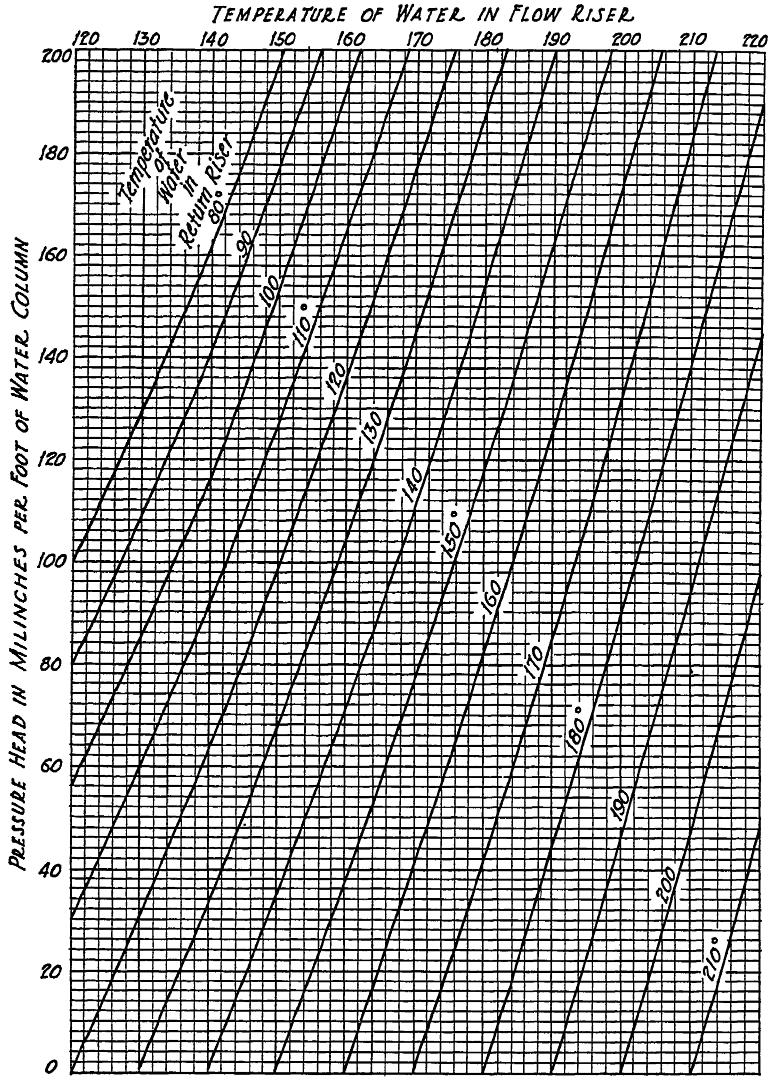


FIG. 6. GRAVITY PRESSURE HEADS FOR VARIOUS TEMPERATURE DIFFERENCES

Since the first section of riser No. 1 is ¾ in. pipe and supplies 28.8 Mbh, it may be noted from Table 2 that a corresponding velocity is approximately 22 in. per second.

From Table 6 a ¾ in. pipe with a velocity of 24 in. per second, used with a 0.35 orifice will produce a loss of 47,000 milinches. For a velocity of 22 in. per second the loss of

head will be less, probably about 41,700 milinches, which is approximately 10 per cent more than the required resistance. This is permissible and the 0.35 in. orifice is selected.

The sizes of the orifice resistors for the second, third and fourth risers are selected in a similar manner and found to be 0.38, 0.42 and 0.50 in. respectively.

GRAVITY CIRCULATION

In a gravity system the motive force to supply circulation is the difference in the weight of the water in the supply and the return and is proportional to the height of the risers. In this system, two distinct heads are available, the head provided in the mains by their elevation above the boiler and the head produced by the elevation of the risers above the mains. From Fig. 6 it is possible to determine the head produced per foot of height by the temperature difference to be used in designing the system. A chart such as Table 1 can be arranged using Fig. 4 for black iron.

To affect a balanced circulation in a gravity hot water heating system careful consideration must be given in sizing the pipes against the amounts of water to be carried, and the head available. The larger the temperature drop, the greater the motive force available.

It is generally customary to use a heat emission of 150 Btu per square foot of radiation, which normally requires an average water temperature of 170 F in the radiator. This can be accomplished by using a 35 F drop with the water entering the radiation at 187 F and leaving at 153 F. Raising the water temperature leaving the boiler will increase the average radiator temperature and alter the heat emission of the radiator.

Assuming that the height of mains above the boiler is 4 ft and that a 35 F drop is desirable, it will be noted that from Fig. 6, a maximum temperature of 200 F and return temperature of 165 F with a pressure head of 150 milinches per foot of height will be produced. A total head of 600 milinches or 0.6 in. is thus produced in the mains. Assuming that the average height of first floor radiators to be 3 ft above the main and second floor radiators to be 12 ft, third floor radiators 21 ft and fourth floor radiators 30 ft, the circulating head will be respectively, 450, 1800, 3150 and 4500 milinches.

The data given in Fig. 4 are based on a 20 F temperature drop which may be converted for capacities of 35 F drop by multiplying the capacity by 1.75. From these data, Tables 7 and 8 may be constructed.

The most common piping layouts used in gravity design are the one-pipe system of Fig. 7 and the two-pipe system of Fig. 8. The same objections are to be found with direct return design in gravity as in forced circulation and the reverse return system of Fig. 2 is to be preferred.

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 7 Column 2, a 2 in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 7 may be used to determine the size of the mains. Note from Column 8, for a 200 ft length, that a 2 in. main will supply 48 Mbh and a 2½ in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2 in. pipe is too small and a 2½ in. pipe

too large. The solution is to use some 2 in. and some 2½ in. pipe. Since the 2½ in. is nearer the correct size than the 2 in., select 2 in. pipe for the first 50 or 60 ft from the boiler and 2½ in. for the remaining pipe back to the boiler.

Tables 8 and 9 may be used to design the radiator risers and connections. According to Table 8, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 9, both radiator tappings should be 1 in. To simplify the construction, select 1 in. flow risers with 1 in. riser branches and 1 in. radiator tappings. Also select 1¼ in. return risers with 1¼ in. riser branches, and 1¼ in. radiator tappings. Similarly, for 18 Mbh, select 1¼ in. flow and return risers and riser branches, and 1¼ in. radiator tappings.

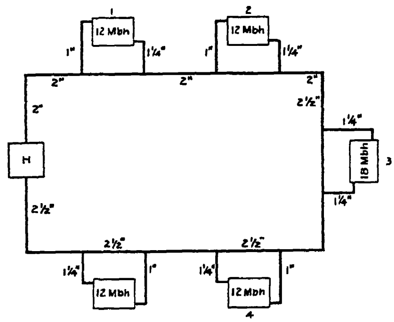


FIG. 7. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

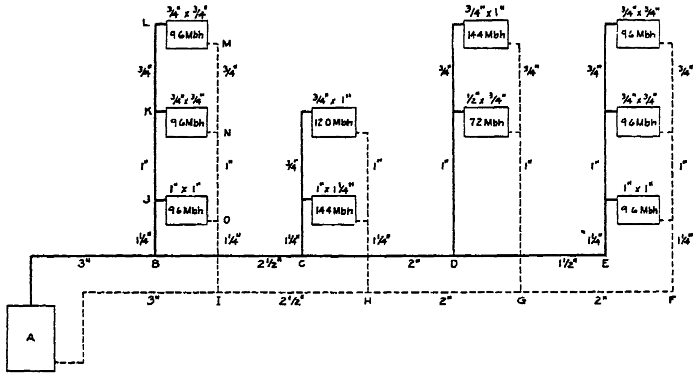


FIG. 8. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 7, in which the total temperature drop is to be 35 F and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the water in the seventh radiator will be 170 F, and, according to Table 4, Chapter 14, the heat dissipation of these two radiators will be to each other as 1.62 is to 1.15, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

TABLE 7. CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F, WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[a])	Equivalent Total Length of Pipe in Feet in Longest Circuit								
		75	100	125	150	175	200	250	300	350
		Unit Friction Head, in Mil-inches								
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7
1½	3.0	43.0	37.5	33.0	30.0	27.0	25.0	22.2	20.2	18.7
2	4.0	83.0	72.0	63.0	57.0	51.0	48.0	42.0	38.0	35.0
2½	4.5	140.0	115.0	100.0	90.0	81.5	75.4	67.2	61.0	56.0
3	5.0	234.0	204.0	175.5	160.0	143.0	133.0	110.0	107.5	100.0
3½	5.5	347.0	300.0	260.0	236.0	214.0	200.0	177.0	160.0	146.0
4	6.0	490.0	422.0	370.0	334.0	297.0	278.0	248.0	223.0	205.0

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 8. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in., the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in. of water.

Table 7 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3 in. pipe is too large and a 2½ in. pipe is too small; hence, select 2½ in. rather than 3 in. as noted in Fig. 8 for Section *AB* and 3 in. for Section *IA*. For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½ in. pipe is almost exactly the correct size and is selected for both sections.

Tables 7 and 8 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 8 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh. According to the table, 1¼ in. flow and return risers will supply 26.0 Mbh; if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and 1¼ in. flow and return risers are selected. However, it must be remembered that the riser branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1 in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a ½ in. flow and a ¾ in. return riser is used, the capacity will be 8.0 Mbh; if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾ in. pipe is selected for both risers.

To design the radiator connections, use Table 9 and note that for the first floor radiator connections the capacity of a ¾ in. flow and 1 in. return is 9.1 Mbh, and that of

TABLE 8. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND Velocities of Water in Pipes in Inches Per Second FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

PIPE SIZE (INCHES)		EQUIVALENT LENGTH OF PIPE (FEET[c])	1ST FLOOR[b]			2ND FLOOR	3RD AND 4TH FLOORS
				Vel. (In. per Sec.[d])			
Flow	Return		*Mbh*	Flow	Return	*Mbh*	*Mbh*
½	½	1.0				5	6.2
½	¾					6.4	8.0
¾	¾	1.5	9	2.3	2.3	10.1	14.0
¾	1		12	3.2	2.0	12.8	17.1
1	1	2.0	18	2.5	2.5	20	26.0
1	1¼		21	3.0	2.0	25.2	34
1¼	1¼	3.0	26	3.0	3.0	43	55
1¼	1½		34	4.0	2.5		
1½	1½	3.5	48	3.0	3.0		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.
[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.
[c]Approximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.
[d]Velocities apply to the riser branches.

a 1 in. flow and a 1 in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1 in. flow and return connection is selected. For the two upper floors, the capacity of a ¾ in. flow and return connection is 10.5 Mbh, and that size is used.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 8 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (see Table 7, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess

TABLE 9. MAXIMUM CAPACITIES OF RADIATOR CONNECTIONS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TEMPERATURE DROP OF 35 F THROUGH EACH RADIATOR

PIPE SIZE		EQUIVALENT LENGTH OF PIPE (FEET[a])	1ST FLOOR	2ND, 3RD, AND 4TH FLOORS
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	4.1	5.9
½	¾		5.2	7.5
¾	¾	1.5	7.0	10.5
¾	1		9.1	13.0
1	1	2.0	12.5	17.8
1	1¼		17.5	23.2
1¼	1¼	3.0	23.3	33.2

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

of that available for the fourth set. The velocity in the riser branch is 3 in. per second (see Table 8) and, therefore, according to Table 6, an 0.65 in. orifice in a 1¼ in. union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70 in. orifice in a 1¼ in. union will provide a resistance of 285 milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60 in. orifice in a 1 in. union will provide sufficient resistance.

EXPANSION TANKS

When water at ordinary temperatures is heated or cooled, its volume is increased or decreased. This variation in the volume of the water in a heating system is generally provided for by means of an expansion tank

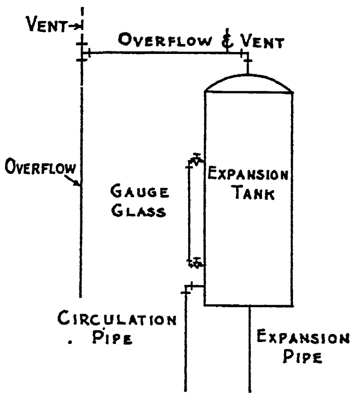


FIG. 9. AN OPEN EXPANSION TANK

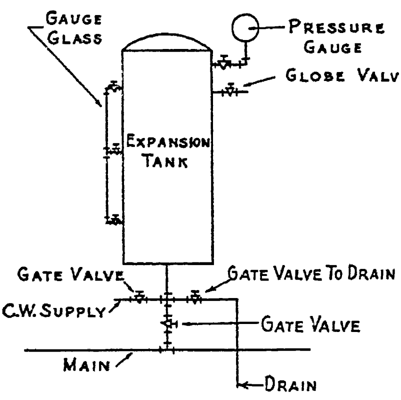


FIG. 10. A CLOSED EXPANSION TANK

into which the water can flow from the system during the heating-up periods and from which it can flow back into the system during the cooling-down periods.

The expansion tank may be open or closed. In an open expansion tank (Fig. 9), the water is subjected to atmospheric pressure and can expand freely without a material increase in pressure. In a closed expansion tank (Fig. 10), the water is subjected to the pressure of the compressed air within the tank, and as the water expands, the volume of the air in the tank is decreased and its pressure increased.

The open expansion tank must be placed at a sufficient elevation above the highest radiator to prevent boiling when the water in that radiator is at the highest temperature to which it is to be heated. For example, if the water is to be heated to 225 F on extremely cold days, the absolute pressure on the water in the highest radiator must be at least 19 lb per square inch. This pressure will be secured if the open expansion tank is located 15 ft above the highest radiator. If a closed expansion tank is used and is located 30 ft below the highest radiator, an absolute pressure

of about 32 lb per square inch must be maintained in the expansion tank if the water in the highest radiator is to be heated to 225 F without danger of boiling.

The type of expansion tank used in a heating system, whether open or closed, has no influence on the operation of the system. The only function performed by the expansion tank is to provide for the variation in the volume of the water in the system, and at the same time to maintain a sufficient pressure in the system to prevent boiling when the water is at the highest temperature for which the system is designed. The capacity of the cushion or expansion tank should not be less than the tank sizes indicated in Table 9 and in addition provisions must be made for draining it without emptying the system.

The capacity of the expansion tank should be at least twice the increase in volume produced when the water in the system is heated from its normal to its maximum temperature. When 25 gal of water are heated

TABLE 9. EXPANSION TANK SIZES FOR HOT WATER HEATING SYSTEMS

TANK SIZE GALLONS	EQUIVALENT DIRECT RADIATION INSTALLED IN SQ FT	CAPACITY DIRECT RADIATION INSTALLED IN MBH
18	Up to 350	Up to 52.5
21	Up to 450	Up to 67.5
24	Up to 650	Up to 97.5
30	Up to 900	Up to 135.0
35	Up to 1100	Up to 165.0
40	Up to 1400	Up to 210.0
2–30	Up to 1600	Up to 240.0
2–30	Up to 1800	Up to 270.0
2–35	Up to 2000	Up to 300.0
2–40	Up to 2400	Up to 360.0

from 40 F to 200 F, the volume of water increases to 26 gal. A safe rule, therefore, is to make the water capacity of the expansion tank equal to 10 per cent of the capacity of the heating system.

In a forced circulation system, the expansion tank can either be connected to the flow or return main. In a gravity circulation system, the expansion tank should be connected to the flow riser so that air liberated from the water in the boiler may escape through the expansion tank.

The expansion tank should be protected so that the water in the tank or in the connecting pipe lines cannot freeze. If the water should freeze and the water in the system is heated causing further expansion, the resulting force will burst the boiler or some other portion of the system.

RELIEF VALVES

A relief valve should be installed on any hot water system using a closed circuit. The valve should be of ample capacity to provide for relief of expansion of the system without allowing an excessive pressure rise above the valve setting.

A relief valve should be of the diaphragm-operated or gravity-weighted type without guide wings below the seat. Provision should be made for manual operation to assure that the valve is in the proper operating condition at all times, and valves should be checked periodically.

A relief valve installed in conjunction with a compression tank will not operate often provided the tank is of adequate size. It is essential that the relief valve be kept in good condition to eliminate any possible failure when operation is necessary.

INSTALLATION DETAILS

The detailed installation of the pipe system should be governed by four fundamental rules:

1. All piping must be pitched either up or down so that all gases which are liberated from the water can move freely to a vented section of the system. Whenever practicable, the pipe line should be pitched so that gases flowing to a vent will flow in the same direction as the water. When a pipe system cannot be installed without creating *air pockets*, that is, sections in the system from which liberated gases cannot escape, such sections must be provided with automatic air relief valves or with air valves which may be operated manually when necessary, or trapped into a pressure tank.

2. All piping must be arranged so that the entire system can be drained, either to permit alterations or repairs, or to prevent freezing if the system is not to be operated during a cold period.

It is well to install a gate valve and union in every riser near the main to permit the draining of individual risers without draining the entire system. It is also well, in large installations, to divide the system into branches and to provide each branch with unions and valves so that any one branch can be drained without disturbing the remaining ones.

The dividing of large heating systems into branches or zones and providing each zone with individual valves has the further advantage of permitting a varying temperature control. For example, if a building is equipped with a forced circulating system and if the south rooms are on one branch of the main and the north rooms are on a separate branch, the valves may be set so that the water will circulate through the north branch with a temperature drop of, say, 10 F, and through the south branch with a temperature drop of, say, 20 F, thus delivering less heat to the south rooms than to the north rooms. This arrangement is especially valuable when the regulating valves are controlled thermostatically by the temperatures in the two zones, because no matter how accurately the heating system may have been designed, the heat demand of any group of rooms varies with sunshine and with wind velocity, and these intermittent variations can be provided for only by the individual control made possible by changing the valve settings controlling the heat supplied to particular groups of rooms.

3. All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

4. The pipe system must be installed so that each circuit has its correct friction head. To bring this about, it is necessary in some cases to minimize the friction, *i.e.*, to make the pipe line as short as possible and to provide as few fittings as possible; and in other cases it is necessary to increase the length of the pipe and the number of fittings so that, for every circuit, the friction head will be equal to the available pressure head.

The connections from the boiler to the mains should be short and direct, to reduce the friction head. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft. The flow main should always be covered; the return main should be covered except where it is to provide the heating surface for the basement.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow

connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side, but in most cases both connections should be at an angle of 45 deg. This method shortens the lines and substitutes 45-deg ells for 90-deg ells.

Preferably, connection of the flow riser to a radiator should be to the upper tapping, and connection of the return riser to a radiator should be to the lower tapping. When hot water enters at the top of a radiator it will distribute itself along the entire length of the radiator, and as it cools it will settle gradually to the bottom; the cool water may then be taken out of the radiator at either end.

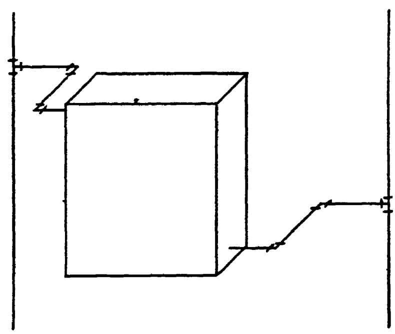


FIG. 11. METHOD OF CONNECTING RADIATOR TO ALLOW FOR EXPANSION OF PIPE

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation and low velocities it makes little difference whether the water leaves at the end at which it enters or at the opposite end.

The connections of the risers to the radiators should be such that provision is made for the vertical expansion of the risers. This can be accomplished as indicated in Fig. 11 by using one tee and two ells for each connection. These connections should be pitched upward or downward, whichever may be necessary to prevent the formation of air pockets and to permit draining.

PROBLEMS IN PRACTICE

1 ● Will altering a hot water heating system from an open to closed type system (*a*) increase the circulation and (*b*) give more heat?

a. No. Tests conducted by the A.S.H.V.E. indicate that there is little, if any difference in the circulation when the system is under pressure. The difference in temperature between the supply and return, and the friction are the governing factors.

b. With a closed system the water may be carried at a higher temperature without boiling which permits warmer radiators.

2 ● What tends to prevent or to retard the circulation of water in hot water heating systems?

In both gravity flow and forced circulation systems, the friction which must be overcome when the water is flowing through pipes, fittings, valves, heaters, and radiators tends to

prevent or retard circulation. For a given pipe the friction varies approximately as the 1.7 power of the velocity, and for given fittings, valves, heaters, and radiators, the friction varies approximately as the square of the velocity. It is therefore sufficiently accurate to express the friction in fittings, valves, heaters, and radiators in terms of the friction in one standard elbow, as shown in Table 3.

3 ● If a single radiator located 10 ft above a boiler is connected with a flow and return black iron pipe, what is the pressure head maintaining the circulation if the water in the return riser is at 180 F and that in the flow riser is at 200 F?

It is found, from Table 7, Chapter 1, that 180 F water weighs 60.61 lb per cubic foot and 200 F water weighs 60.13 lb per cubic foot. The pressure head is independent of the size of the pipe. If the two risers were each 1 ft square, the water in the flow riser would weigh 601.3 lb and that in the return riser would weigh 606.1 lb. Thus the water in the return riser would weigh 4.8 lb more than that in the flow riser. Consequently, the resulting pressure head is 4.8 lb per square foot.

Pressure heads are generally expressed in feet, or inches, or milinches of water of a given temperature. In this case water is at both 180 F and 200 F, so the pressure head is expressed in terms of 190 F water. Such water weighs 60.39 lb per cubic foot, and to secure a pressure of 4.8 lb per square foot, it is necessary to have a column of water having a weight of 4.8 divided by 60.39 = 0.0795 ft, or 0.9540 in., or 954 milinches. This is the pressure head which maintains the circulation.

4 ● In the elementary system of Question 3, if the radiator dissipates 14,000 Btu per hour, what is the velocity of the water in the pipe line, if the pipes are 1 in. in diameter? What, if they are ¾ in. in diameter?

Since the temperature drop through the radiator is from 200 F to 180 F or 20 F, every pound of water flowing through the radiators delivers 20 Btu; consequently, 14,000 divided by 20 = 700 lb of water, or for 190 F water, 700 divided by 60.39 = 11.59 cu ft of water must flow through the radiator and through the pipe lines every hour.

The interior area of a 1 in. pipe is 0.864 sq in. The velocity in the 1 in. pipe is 11.59 divided by 0.864 and multiplied by 144 = 1932 ft per hour or 6.44 in. per second.

For ¾ in. pipe, the interior area is 0.533, and the velocity is 6.44 multiplied by 0.864 and divided by 533 = 10.44 in. per second.

5 ● If, in the elementary heating system of Question 3, a 1 in. pipe line is used, what would be the friction head?

If the radiator is connected with the heater to provide for freedom of expansion, the heating circuit may be assumed to consist of a heater, 25 ft of pipe, 8 elbows, 1 radiator valve, and 1 radiator. From Table 3 it appears that the heater and radiator are equivalent, in friction, to 6 elbows; hence, the circuit may be placed equal to 25 ft of pipe and 14 elbows.

From the diagram of Fig. 4 it appears that the friction head for a 1 in. pipe and a velocity of 6.44 in. per second is about 25 milinches per foot. For 25 ft of pipe, the friction head will be 625 milinches.

It appears from Table 3 that the friction head in one elbow is $\frac{v^2}{2g}$, or in this case 0.54 multiplied by 0.54 and divided by 64.4 = 0.0045 ft or 54 milinches. Hence, for the 14 elbows the friction is 756 milinches. For the entire circuit, the friction head is the sum of the 625 milinches of the pipe plus the 756 milinches of the elbows, or 1381 milinches which equal 1.381 in.

6 ● If the elementary heating system of Question 3 is installed with a 1 in. pipe line, how will it function?

It is found from the answer to Question 3 that the pressure head is 954 milinches and from the answer to Question 5 that the friction head is 1381 milinches when the water is flowing with such velocity that the specified 14,000 Btu will be delivered with a 20 F temperature drop through the radiators. Since the pressure head is smaller than the friction head, the system will not function as planned for the water will flow through the

system more slowly and remain in the radiator longer. The temperature drop through the radiator will be more than 20 F, and the difference in the weight of the water in the return and flow risers will be greater than that intended. The final result will be that the pressure head will become equal to the friction head at a value somewhere between 954 and 1381 milinches. Since the average water temperature in the radiator will be less than 190 F, the radiator should be larger than the size given in Question 4.

7 ● Should a hot water heating system be designed to embody small pipes or large pipes?

As pipe sizes in gravity circulation heating are reduced, the friction head is increased and it is necessary to increase the temperature drop through radiators; this lowers the average temperature of the water in the radiators and necessitates an increase in the size of the radiators, so whereas the cost of the pipe in a system is reduced, the cost of the radiators is increased. For each installation there is a definite pipe size which entails maximum economy.

As pipe sizes in forced circulation systems are reduced, friction heads are increased so a circulating pump of greater size or capacity is required. Thus, by decreasing the size of the piping, both the first cost of the circulating pump and the cost of its operation are increased. There is a definite pipe size for every installation which is most economical. For each installation of both types of systems there is a definite pipe size entailing maximum economy which can be determined by a series of comparative calculations.

8 ● What should be the size of the radiators for the elementary heating system of Question 3 in which the water enters the radiator with a temperature of 200 F and leaves with a temperature of 180 F? The average temperature of the water in the radiator is, approximately, 190 F.

If test results are available for the particular radiators to be used, and for the temperatures named, the size of the radiators should be selected from them. If no such test results are to be had, but if test results are available for the type of radiator to be used when it is supplied with 215 F steam and placed in a 70 F room, the required size may be determined by the following ratio: The required size is to the corresponding steam radiator size as $(215 - 70)^{1.3}$ is to $(190 - 70)^{1.3}$. This ratio works out to 1.28. Hence, the radiators should be 28 per cent larger under the conditions prescribed than are corresponding radiators under standard conditions. It is immaterial whether a radiator is filled with steam or with water, as long as the average temperature of its outer surface is the same in both cases.

Chapter 18

PIPE, FITTINGS, WELDING

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe.*

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld or butt-weld process. While the lap-weld process produces a better weld than the butt type, lap-weld pipe is seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is considered to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher

first cost can be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast Ferrous Pipe. There are now available several types of cast ferrous-metal pipe made of a good grade of cast iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra-strong wrought pipe. Cast ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin-wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall

thicknesses listed. In sizes 14 in. and upward, pipe is designated by it outside diameter (O.D.) and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry ar reasonably well served by the *standard-weight* and *extra-strong* pipe demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Eve in heating installations, the erection of piping by welding was deemed t

TABLE 1. DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE

NOMINAL PIPE SIZE	OUTSIDE DIAM.	NOMINAL WALL THICKNESSES FOR SCHEDULE NUMBERS									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
1/8	0.405				0.068*		0.095*				
1/4	0.540				0.088*		0.119*				
3/8	0.675				0.091*		0.126*				
1/2	0.840				0.109*		0.147*				0.18
3/4	1.050				0.113*		0.154*				0.21
1	1.315				0.133*		0.179*				0.25
1¼	1.660				0.140*		0.191*				0.25
1½	1.900				0.145*		0.200*				0.28
2	2.375				0.154*		0.218*				0.34
2½	2.875				0.203*		0.276*				0.37
3	3.500				0.216*		0.300*				0.43
3½	4.000				0.226*		0.318*				
4	4.500				0.237*		0.337*		0.437		0.53
5	5.563				0.258*		0.375*		0.500		0.62
6	6.625				0.280*		0.432*		0.562		0.71
8	8.625		0.250	0.277*	0.322*	0.406	0.500*	0.593	0.718	0.812	0.90
10	10.75		0.250	0.307*	0.365*	0.500*	0.593	0.718	0.843	1.000	1.12
12	12.75		0.250	0.330*	0.406	0.562	0.687	0.843	1.000	1.125	1.31
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.40
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.56
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.75
20 O. D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.93
24 O. D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.31
30 O. D.	30.0	0.312	0.500	0.625							

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wa dimensions and include an allowance for mill tolerance of 12.5 per cent under nominal thicknesses.

*Thicknesses marked with asterisk in Schedules 30 and 40 are identical with thicknesses for *standard weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-stron* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

warrant the use of pipe lighter than standard weight. For these reasons a *Sectional Committee on Standardization of Wrought Iron and Wrough Steel Pipe and Tubing* functioning under the procedure of the *America Standards Association* was appointed to standardize the dimensions an materials of pipe.

The proposed pipe standard recommended by that sectional committe has set up several schedules of pipe including standard-weight and extra strong thicknesses which are now included in Schedules 40 and 60, re spectively. The schedules approved by the Sectional Committee ar given in Tables 1 and 3 and the corresponding weights in Tables 2 and 4

Standard-weight pipe is generally furnished with threaded ends i

random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials.* There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Class *K*—Designed for underground services and general plumbing service.

Class *L*—Designed for general plumbing purposes.

Class *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and

TABLE 2. NOMINAL WEIGHTS OF WELDED AND SEAMLESS STEEL PIPE

Nominal Pipe Size Inches	Sched. 10 Plain Ends	Sched. 20 Plain Ends	Schedule 30		Schedule 40		Sched. 60 Plain Ends	Sched. 80 Plain Ends	Sched. 100 Plain Ends	Sched. 120 Plain Ends	Sched. 140 Plain Ends	Sched 160 Plain Ends
			Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings						
⅛					0.25*	0.25*		0.32*				
¼					0.43*	0.43*		0.54*				
⅜					0.57*	0.57*		0.74*				
½					0.86*	0.86*		1.09*				1.3
¾					1.14*	1.14*		1.48*				1.9
1					1.68*	1.69*		2.18*				2.8
1¼					2.28*	2.29*		3.00*				3.7
1½					2.72*	2.74*		3.64*				4.8
2					3.66*	3.68*		5.03*				7.4
2½					5.80*	5.82*		7.67*				10.0
3					7.58*	7.62*		10.3*				14.3
3½					9.11*	9.21*		12.5*				
4					10.8*	10.9*		15.0*		19.0		22.6
5					14.7*	14.9*		20.8*		27.1		33.0
6					19.0*	19.2*		28.6*		36.4		45.3
8		22.4	24.7*	25.0*	28.6*	28.8*	35.7	43.4*	50.9	60.7	67.8	74.7
10		28.1	34.3*	35.0*	40.5*	41.2*	54.8*	64.4	77.0	89.2	105.0	116.0
12		33.4	43.8*	45.0*	53.6	55.0	73.2	88.6	108.0	126.0	140.0	161.0
14 O. D.	36.8	45.7	54.6		63.3		85.0	107.0	131.0	147.0	171.0	190.0
16 O. D.	42.1	52.3	62.6		82.8		108.0	137.0	165.0	193.0	224.0	241.0
18 O. D.	47.4	59.0	82.0		105.0		133.0	171.0	208.0	239.0	275.0	304.0
20 O. D.	52.8	78.6	105.0		123.0		167.0	209.0	251.0	297.0	342.0	374.0
24 O. D.	63.5	94.7	141.0		171.0		231.0	297.0	361.0	416.0	484.0	536.0
30 O. D.	99.0	158.0	197.0									

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which ar commercially available with threads and couplings for which both weights are listed.

*The weights marked with asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe i former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

Types L and M where such conditions may be considered normal as, for instance, in heating work. Types K and L are available in both hard and soft tempers; Type M is available only in hard temper. Where flexibility is essential as in hidden replacement work or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. New or exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall thickness tolerances for these classes of copper tubing are given in Table 5. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 F or more above room tem-

TABLE 3. DIMENSIONS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size	Outside Diameter	Nominal Wall Thicknesses for Schedule Numbers					
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80
⅛	0.405	....	....	....	0.070*	....	0.098*
¼	0.540	....	....	....	0.090*	....	0.122*
⅜	0.675	....	....	....	0.093*	....	0.129*
½	0.840	....	....	....	0.111*	....	0.151*
¾	1.050	....	....	....	0.115*	....	0.157*
1	1.315	....	....	....	0.136*	....	0.183*
1¼	1.660	....	....	....	0.143*	....	0.195*
1½	1.900	....	....	....	0.148*	....	0.204*
2	2.375	....	....	....	0.158*	....	0.223*
2½	2.875	....	....	....	0.208*	....	0.282*
3	3.5	....	....	....	0.221*	....	0.306*
3½	4.0	....	....	....	0.231*	....	0.325*
4	4.5	....	....	....	0.242*	....	0.344*
5	5.563	....	....	....	0.263*	....	0.383*
6	6.625	....	....	....	0.286*	....	0.441*
8	8.625	....	....	0.283*	0.329*	....	0.510*
10	10.75	....	....	0.313*	0.372*	0.510*	0.606
12	12.75	....	....	0.336*	0.414	0.574	0.702
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.625	0.750
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.687	....
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.750	....
20 O. D.	20.0	....	0.375	0.500	0.562	....	....

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under the nominal thickness.

*Thicknesses marked with an asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P S

perature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast iron, steel, wrought iron, and copper pipe,

TABLE 4. NOMINAL WEIGHTS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size (Inches)	Sched. 10	Sched. 20	Schedule 30		Schedule 40		Schedule 60	Schedule 80
	Plain Ends	Plain Ends	Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings	Plain Ends	Plain Ends
1/8	....	....	....	....	0.25*	0.25*	....	0.32*
1/4	....	....	....	....	0.43*	0.43*	....	0.54*
3/8	....	....	....	....	0.57*	0.57*	....	0.74*
1/2	....	....	....	....	0.86*	0.86*	....	1.09*
3/4	....	....	....	....	1.14*	1.14*	....	1.48*
1	....	....	....	....	1.68*	1.69*	....	2.18*
1 1/4	....	....	....	....	2.28*	2.29*	....	3.00*
1 1/2	....	....	....	....	2.72*	2.74*	....	3.64*
2	....	....	....	....	3.66*	3.68*	....	5.03*
2 1/2	....	....	....	....	5.80*	5.82*	....	7.67*
3	....	....	....	....	7.58*	7.62*	....	10.3*
3 1/2	....	....	....	....	9.11*	9.21*	....	12.5*
4	....	....	....	....	10.8*	10.9*	....	15.0*
5	....	....	....	....	14.7*	14.9*	....	20.8*
6	....	....	....	....	19.0*	19.2*	....	28.6*
8	....	....	24.7*	25.0*	28.6*	28.8*	....	43.4*
10	....	....	34.3*	35.0*	40.5*	41.2*	54.8*	54.4
12	....	....	43.8*	45.0*	53.6	55.0	73.2	88.6
14 O. D.	36.0	44.8	53.6	....	62.2	....	87.6	104.0
16 O. D.	41.3	51.4	61.4	....	81.2	....	111.0	....
18 O. D.	46.5	57.9	80.5	....	103.0	....	136.0	....
20 O. D.	....	77.0	103.0	....	115.0	....	....	....

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*Weights marked with an asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

the materials most frequently used in heating and ventilating work, can be determined from Table 6.

The elongation values in Table 6 were computed from the following formula:

$$L_t = L_o \left[1 + a\left(\frac{t - 32}{1000}\right) + b\left(\frac{t - 32}{1000}\right)^2\right] \quad (1)$$

where

L_t = length at temperature t degrees Fahrenheit, feet.
L_o = length at 32 F, feet.
t = final temperature, degrees Fahrenheit.
a and b are constants as given on the next page.

METAL	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought-Iron	0.006503	0.001622
Copper	0.009278	0.001244

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

TABLE 5. STANDARD DIMENSIONS, WEIGHTS, AND DIAMETER AND WALL THICKNESS TOLERANCES FOR COPPER WATER TUBES*

(All Tolerances Plus and Minus)

NOMINAL SIZE, IN.	ACTUAL OUTSIDE DIAMETER, IN.	PERMISSIBLE VARIATION IN MEAN OUTSIDE DIAMETER, IN.		WALL THICKNESS, IN.						WEIGHT PER FT LB		
				Class *K*		Class *L*		Class *M*				
		Annealed	Hard Drawn	Nominal	Permissible Variation	Nominal	Permissible Variation	Nominal	Permissible Variation	Class *K*	Class *L*	Class *M*
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.144
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.203
3/4	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.464
1 1/4	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.681
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.94
2	2.125	0.005	0.002	0.083	0.005	0.070	0.005	0.058	0.0045	2.06	1.75	1.46
2 1/2	2.625	0.005	0.002	0.095	0.005	0.080	0.005	0.065	0.0045	2.92	2.48	2.03
3	3.125	0.005	0.002	0.109	0.005	0.090	0.005	0.072	0.0045	4.00	3.33	2.68
3 1/2	3.625	0.005	0.002	0.120	0.005	0.100	0.005	0.083	0.005	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.006	0.110	0.005	0.095	0.005	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.006	0.125	0.006	0.109	0.005	9.67	7.61	6.65
6	6.125	0.005	0.002	0.192	0.006	0.140	0.006	0.122	0.005	13.87	10.20	8.91

*From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials, A.S.T.M.* Designation B88-33.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since con-

tinued turning in the threaded joint may in time result in a leak, particularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion

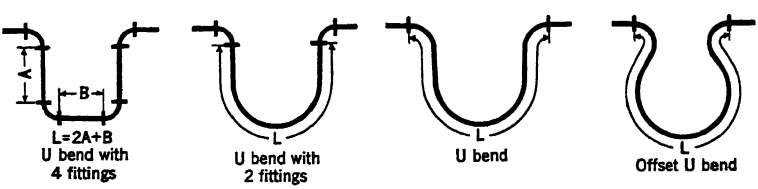


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from the following formula:

$$L = 6.16 \sqrt{D \Delta} \tag{2}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter.

All risers must be anchored and safeguarded so that the difference in

[1]Piping Handbook, by Walker and Crocker, and A Manual for the Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, and E. T. Cope, published by *The Detroit Edison Company.*

length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

It is especially necessary with light-weight radiators so to anchor the piping and so to give it freedom for expansion that no strain therefrom shall be allowed to distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

TABLE 6. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 Ft from −20 F up				Saturated Steam		Elongation in Inches per 100 Ft from −20 F up			
Vacuum Inches of Hg.	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39	....	60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45	....	200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from − 20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast iron, but in addition to this malleable iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another,

may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types is shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in large and small sizes. While no effort has been made to standardize dimensions of flared tube fittings, manufacturers have quite generally used *S.A.E.* standard dimensions. Flared tube fittings are widely used in refrigeration work and the use of *S.A.E.* dimensions and a 45 deg flare renders most fittings

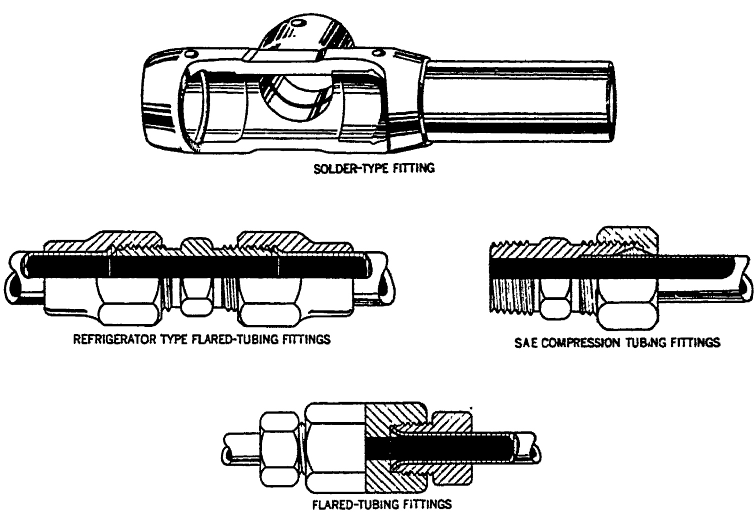


FIG. 2. COPPER OR BRASS TUBING FITTINGS

interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits.

Ammonia pipe fittings made of cast iron are extensively used in handling refrigerants in larger installations. Until recently, no standard dimensions were adhered to in the manufacture of ammonia flanged fittings or companion flanges with the result that fittings of different manufacturers were not interchangeable. A subcommittee of *A.S.A* Sectional Committee B16 has prepared proposed American Standard dimensions for ammonia flanged fittings and companion flanges for maximum service pressure of 300 lb per sq in. which will be available soon.

Thread Connections

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand

threads are used. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$-inch raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a $\frac{1}{16}$-inch raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 7, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 8 and 9.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a relatively narrow recessed facing.

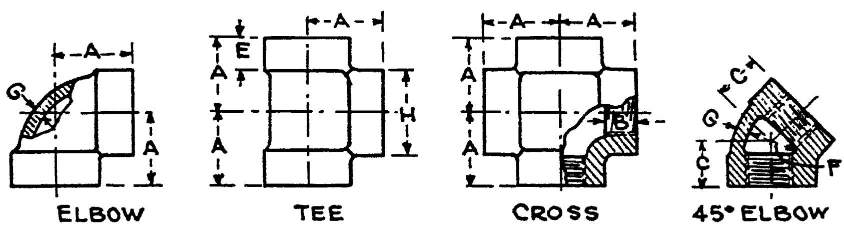
WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as a competitive method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting

TABLE 7. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45 DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON SCREWED FITTINGS



	A	C	B	E	F		G	H
Nominal Pipe Size	Center to End, Elbows, Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread Min.	Width of Band, Min.	Inside Diameter of Fitting		Metal Thickness, Min.	Outside Diameter of Band, Min.
					Min.	Max.		
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50	5.97	1.88	1.88	12.750	12.850	0.800	15.47
14 O.D.	10.40	....	2.00	2.00	14.000	14.100	0.880	16.94
16 O.D.	11.82	....	2.20	2.20	16.000	16.100	1.000	19.30

All dimensions given in inches.

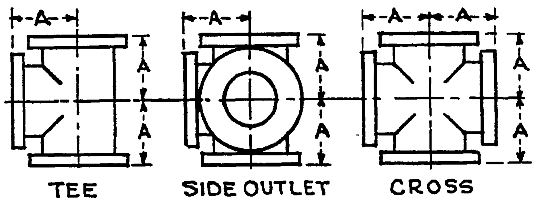
all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing

the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available

TABLE 8. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON FLANGED FITTINGS



Nominal Pipe Size[a-b]	*A* Center to Face Tees and Crosses[b-c]	*AA* Face to Face Tees and Crosses[b-c]	Diameter of Flange	Thickness of Flange, Min.	Metal Thickness of Body, Min.
1	3½	7	4¼	7/16	7/16
1¼	3¾	7½	4⅝	½	7/16
1½	4	8	5	9/16	7/16
2	4½	9	6	⅝	7/16
2½	5	10	7	11/16	7/16
3	5½	11	7½	¾	7/16
3½	6	12	8½	13/16	7/16
4	6½	13	9	15/16	½
5	7½	15	10	15/16	½
6	8	16	11	1	9/16
8	9	18	13½	1⅛	⅝
10	11	22	16	1 3/16	¾
12	12	24	19	1¼	13/16
14 O.D.	14	28	21	1⅜	⅞
16 O.D.	15	30	23½	1 7/16	1
18 O.D.	16½	33	25	1 9/16	1 1/16
20 O.D.	18	36	27½	1 11/16	1⅛
24 O.D.	22	44	32	1⅞	1¼
30 O.D.	25	50	38¾	2⅛	1 7/16
36 O.D.	28	56	46	2⅜	1⅝
42 O.D.	31	62	53	2⅝	1 13/16
48 O.D.	34	68	59½	2¾	2

All dimensions given in inches.

[a]Size of all fittings listed indicates nominal inside diameter of port.

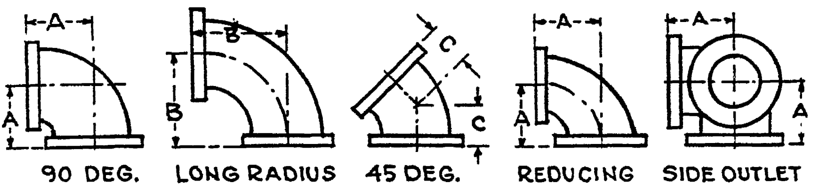
[b]Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[c]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

with plain ends machine beveled for welding and with radii similar to short and long radius flanged fittings. Some typical types of these fittings are shown in Fig. 3. They are made in pipe sizes ¾ to 24 in., standard and extra heavy, in steel, wrought iron, brass, copper, and special alloys.

Socket welding fittings of forged steel are also commercially available. These fittings have a machined recess into which the pipe slips. A fillet weld between the pipe and socket edge provides a pressure-tight joint. A proposed American Standard containing dimensions of steel welding-neck flanges for pressures up to 1500 lb per sq in. has been developed in *A.S.A*

TABLE 9. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR 125 LB CAST-IRON FLANGED FITTINGS



Nominal Pipe Size[a]	A	B	C	Diameter of Flange	Thickness of Flange, Min.	Metal Thickness of Body, Min.
	Center to Face Elbow b-c-d	Center to Face Long Radius Elbow b-c-d	Center to Face 45 Deg Elbow c			
1	3½	5	1¾	4¼	7/16	7/16
1¼	3¾	5½	2	4⅝	½	7/16
1½	4	6	2¼	5	9/16	7/16
2	4½	6½	2½	6	⅝	7/16
2½	5	7	3	7	11/16	7/16
3	5½	7¾	3	7½	¾	7/16
3½	6	8½	3½	8½	13/16	7/16
4	6½	9	4	9	15/16	½
5	7½	10¼	4½	10	15/16	½
6	8	11½	5	11	1	9/16
8	9	14	5½	13½	1⅛	⅝
10	11	16½	6½	16	1 3/16	¾
12	12	19	7½	19	1¼	13/16
14 O.D.	14	21½	7½	21	1⅜	⅞
16 O.D.	15	24	8	23½	1 7/16	1
18 O.D.	16½	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	29	9½	27½	1 11/16	1⅛
24 O.D.	22	34	11	32	1⅞	1¼
30 O.D.	25	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	49	18	46	2⅜	1⅝
42 O.D.	31	56½	21	53	2⅝	1 13/16
48 O.D.	34	64	24	59½	2¾	2

All dimensions given in inches.

[a]Size of all fittings listed indicates nominal inside diameter of port.

[b]Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[c]Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45 deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90 deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

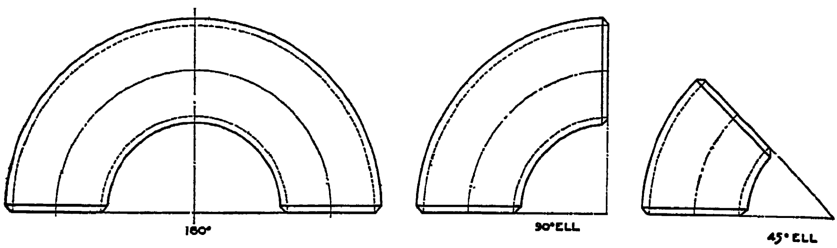
[d]Side outlet elbows shall have all openings on intersection center-lines.

Sectional Committee B16. Tables 10 and 11 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per sq in. gage pressure.

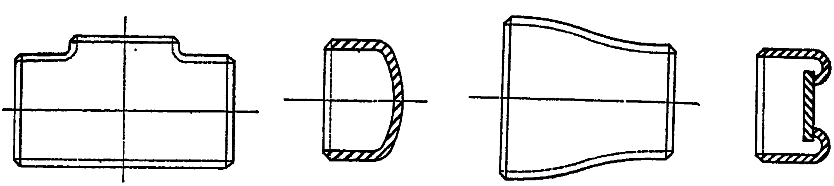
VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed.



a. TYPICAL SHORT RADIUS ELBOWS



b. TEE   *c.* FORGED CAP   *d.* CONCENTRIC REDUCER   *e.* END CLOSURE

FIG. 3. TYPICAL WELDING FITTINGS

Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

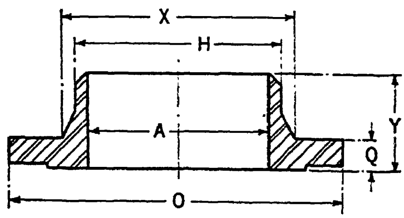
Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum. These valves may be secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether

the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between

TABLE 10. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB AT 750 F



Nominal Pipe Size	Diameter of Flange	Thickness of Flg. Min.	Diameter of Hub	Hub Diam. Beginning of Chamfer	Length Thru Hub	Diam. for Standard Pipe	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	Q	X	H	Y	A			
1	4¼	7/16	1 15/16	1.32	2 3/16	1.05	3⅛	4	½
1¼	4⅝	½	2 5/16	1.66	2¼	1.38	3½	4	½
1½	5	9/16	2 9/16	1.90	2 7/16	1.61	3⅞	4	½
2	6	⅝	3 1/16	2.38	2½	2.07	4¾	4	⅝
2½	7	11/16	3 9/16	2.88	2¾	2.47	5½	4	⅝
3	7½	¾	4¼	3.50	2¾	3.07	6	4	⅝
3½	8½	13/16	4 13/16	4.00	2 13/16	3.55	7	8	⅝
4	9	15/16	5 5/16	4.50	3	4.03	7½	8	⅝
5	10	15/16	6 7/16	5.56	3½	5.05	8½	8	¾
6	11	1	7 9/16	6.63	3½	6.07	9½	8	¾
8	13½	1⅛	9 11/16	8.63	4	7.98	11¾	8	¾
10	16	1 3/16	12	10.75	4	10.02	14¼	12	⅞
12	19	1¼	14⅜	12.75	4½	12.00	17	12	⅞
14 O. D.	21	1⅜	15¾	14.00	5	13.25	18¾	12	1
16 O. D.	23½	1 7/16	18	16.00	5	15.25	21¼	16	1
18 O. D.	25	1 9/16	19⅞	18.00	5½	17.25	22¾	16	1⅛
20 O. D.	27½	1 11/16	22	20.00	5 11/16	19.25	25	20	1⅛
24 O. D.	32	1⅞	26⅛	24.00	6	23.25	29½	20	1¼

All dimensions given in inches.

A raised face of 1/16 in. is included in *thickness of flange minimum*.

It is recommended that the taper of the hub should not exceed 6 deg for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

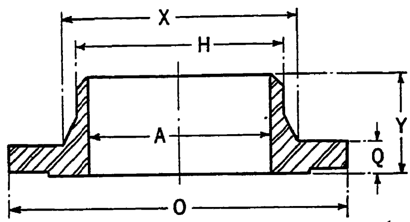
the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever

handles are often supplied to indicate the relative opening of the valve in any position. Standard roughing-in dimensions for angle-type valves are given in Table 12.

Automatic control of steam supply to individual radiators can be

TABLE 11. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 300 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 750 F



Nominal Pipe Size	Diam. of Flange	Thickness of Flange Min.	Diam. of Hub	Hub Diam. Beginning of Chamfer	Length Thru Hub	Diam. for Standard Pipe	Diam. for Extra Strong Pipe	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	*Q*	*X*	*H*	*Y*	*A*	*A*			
*2	6½	⅞	3 5/16	2.38	2¾	2.07	1.94	5	8	⅝
2½	7½	1	3 15/16	2.88	3	2.47	2.32	5⅞	8	¾
3	8¼	1⅛	4⅝	3.50	3⅛	3.07	2.90	6⅝	8	¾
3½	9	1 3/16	5¼	4.00	3 3/16	3.55	3.36	7¼	8	¾
4	10	1¼	5¾	4.50	3⅜	4.03	3.83	7⅞	8	¾
5	11	1⅜	7	5.56	3⅞	5.05	4.81	9¼	8	¾
6	12½	1 7/16	8⅛	6.63	3⅞	6.07	5.76	10⅝	12	¾
8	15	1⅝	10¼	8.63	4⅜	7.98	7.63	13	12	⅞
10	17½	1⅞	12⅝	10.75	4⅝	10.02	9.75	15¼	16	1
12	20½	2	14¾	12.75	5⅛	12.00	11.75	17¾	16	1⅛
14 O. D.	23	2⅛	16¾	14.00	5⅝	13.25	....	20¼	20	1⅛
16 O. D.	25½	2¼	19	16.00	5¾	15.25	....	22½	20	1¼
18 O. D.	28	2⅜	21	18.00	6¼	17.25	....	24¾	24	1¼
20 O. D.	30½	2½	23⅛	20.00	6⅜	19.25	....	27	24	1¼
24 O. D.	36	2¾	27⅝	24.00	6⅝	23.25	....	32	24	1½

*For sizes below 2 in. use dimensions of 600 lb flanges.

All dimensions given in inches.

A raised face of 1/16 in. is included in *thickness of flange minimum*.

It is recommended that the taper of the hub should not exceed 6 deg for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

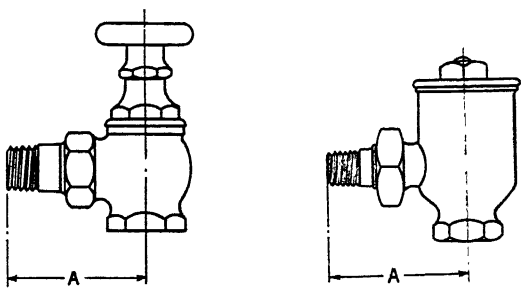
effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a 1/16-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly

for use in hot water heating systems are of less complex design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when

TABLE 12. STANDARD ROUGHING-IN DIMENSIONS ANGLE TYPE VALVES



SIZE OF VALVE	DIMENSION A STEAM AND HOT WATER ANGLE VALVES AND UNION ELBOWS EFFECTIVE JANUARY 1, 1926	DIMENSION A MODULATING VALVES EFFECTIVE JANUARY 1, 1926	DIMENSION A RETURN LINE VACUUM VALVES EFFECTIVE JANUARY 1, 1925
½	2¼	2¾	3¼
¾	2¾	2¾	------
1	3	3	------
1¼	3½	3½	------
1½	3¾	3¾	------
2	4¼	4¼	------
Tolerance	±⅛	±⅛	------

All dimensions given in inches.

Connecting ends shall be threaded and gaged as to threading according to the American (Taper) Pipe Thread Standard, *A.S.A.* No. B2—1919.

The standardization of the Roughing-in Dimensions of Angle Steam and Hot Water, and Modulating Radiator Valves was made possible by the cooperation of the Manufacturers Standardization Society of the Valves and Fittings Industry.

a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system operating continuously or intermittently and supplied with vacuum valves will generally heat more quickly than one provided with non-vacuum air valves; thus, it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. In those cases, where a system is equipped with vacuum air valves and which has been cold for several days, the system will probably have an

internal pressure within the radiator closely approaching atmospheric. At such times, the vacuum valve will not vent the system any more rapidly than the ordinary type. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION[2]

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion in steam heating systems occurs principally in the condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaeration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible.

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[2]New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933). Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 253). Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 263). Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1937, p. 387).

The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[3]. The exact application of this law, however, assumes equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

Distinction should be made between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam, such as water heaters, kitchen equipment, and sterilizers. Experience has shown that in heating systems the partial pressures of the gases do not reach such magnitudes as to cause harmful amounts of gas to become dissolved in the condensate when steam supplies are of reasonable purity. In other kinds of steam-using apparatus which are not ordinarily well vented, the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations. Consequently, corrosion is frequently observed in the condensate discharge lines from such apparatus, but this does not necessarily indicate that equally serious corrosion is taking place in the heating system supplied with steam from the same source.

When corrosive conditions are believed to exist, their seriousness should be determined by actual measurement, rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosiveness of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaeration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Some success has been reported with the use of inhibitors, chief among which are oil, and sodium silicate. Oil may be fed into the main steam-supply pipe by means of a sight-feed lubricator. The type of oil known as 600-W is usually recommended. In the present state of knowledge on this point, the quantity to be fed can best be determined by trial. The use of sodium silicate, fed in a similar manner, is reported to be successful but it has not been widely used.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to

[3]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 121).

this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

PROBLEMS IN PRACTICE

1 ● What is the meaning of IPS brass pipe?

It means that the brass pipe has the same external diameter as steel pipe in the same nominal pipe size and that the wall thickness is sufficient to allow cutting of threads for use with standard size threaded fittings.

2 ● Why is thin-walled copper pipe made up with sweated joints?

If the pipe were threaded it would be necessary to use at least standard-weight wall thickness on account of the metal removed in threading. Flared ends with coupling nuts may be used, but this construction is expensive and hard to keep tight.

3 ● How are pipes designated in diameters of 12 in. and less?

By weight and nominal size, referring to the approximate inside diameter.

4 ● How are pipe sizes designated in diameters of 14 in. and more?

By wall thickness and outside diameter.

5 ● Why are expansion joints required in steam pipes?

To care for the change in length of the line brought about by a change in temperature.

6 ● What devices are used for taking up expansion?

Expansion joints, swivel joints, and the inherent flexibility of the pipe itself.

7 ● Where are swivel joints principally used?

In branch connections to radiators, and in the risers of multi-story buildings where they are installed between the floor joists.

8 ● Name three grades of American Standard screwed pipe fittings.

125-lb cast-iron, 150-lb malleable iron, and 250-lb cast-iron.

9 ● In what sizes are American Standard cast-iron flanges and flanged fittings for 25-lb saturated steam pressure made?

In nominal sizes from 4 in. to 72 in., inclusive.

10 ● What fittings are generally used for threaded connections in low pressure heating systems?

Cast-iron.

Chapter 19

GRAVITY WARM AIR FURNACE SYSTEMS

Design Procedure, Estimating Heating Requirements, Leader Pipe Sizes, Proportioning Wall Stacks, Register Selections, Recirculating Ducts and Grilles, Furnace Return Connection, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 20. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

DESIGN PROCEDURE

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building.
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers.
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

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[1]All figures and much of the engineering data which follow are from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo.

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This *size* should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 7, taking into consideration the transmission losses as well as the infiltration losses.

LEADER PIPE SIZES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating and Air Conditioning Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \qquad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \qquad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \qquad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \qquad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be thoroughly covered or the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

The values shown in Fig. 1 apply only to the case where the straight, leader pipe is 8 ft in length and is connected to stacks whose cross-sectional area is approximately 75 per cent of that of the leader pipe.

Any deviation from these conditions requires a modification of the constants used in Equations 1, 2, and 3. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. It is not considered good commercial practice to specify diameters except

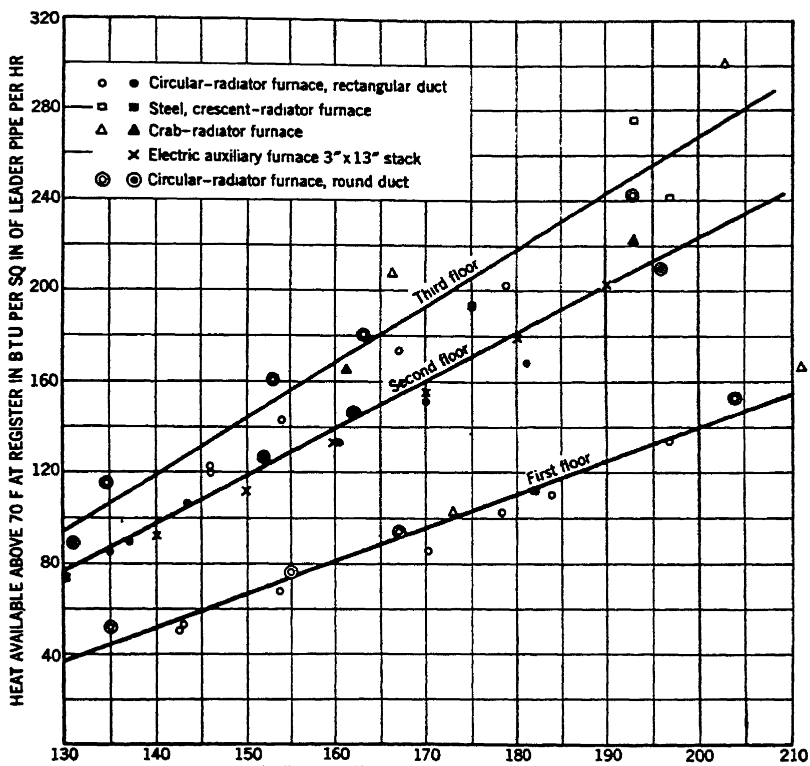

FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length should be avoided if possible. In cases where such leaders are required, the use of a larger size pipe, than is required by the application of the equations, smooth transition fittings, and duct insulation are recommended.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader. In cases where the leader is short and straight as was the case for Fig. 1, such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in

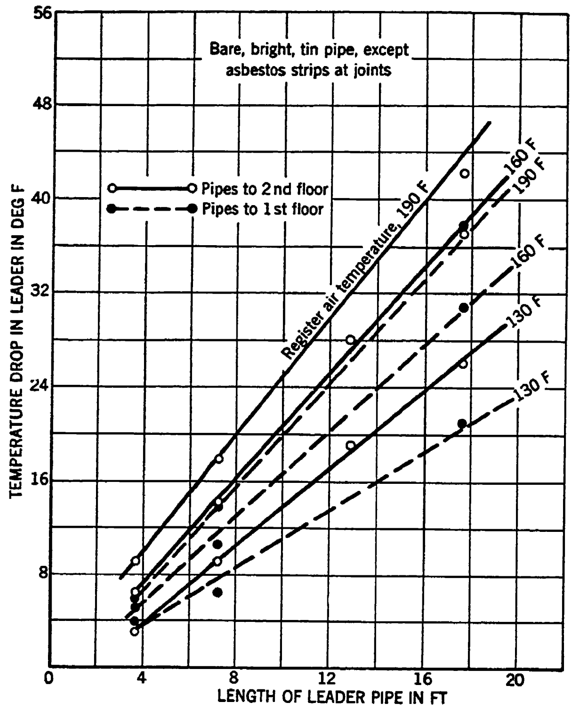

FIG. 2. INFLUENCE OF LEADER PIPE LENGTH ON TEMPERATURE LOSS IN AIR FLOWING THROUGH PIPE

order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this ratio of stack to leader area is a very important matter.

The curves in Figs. 4 and 5 indicate that for rooms having a heat requirement exceeding approximately 9000 Btu per hr, exceedingly high register temperatures are required for stacks whose width is less than 3½ in. For such requirements either multiple stacks, or stacks having larger cross-sectional area (placed in 6 in. studding spaces) will be required.

REGISTER SELECTIONS

The registers used for discharging warm air into the rooms should have a free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First-floor registers may be of the baseboard or floor type, with the former location preferred. High

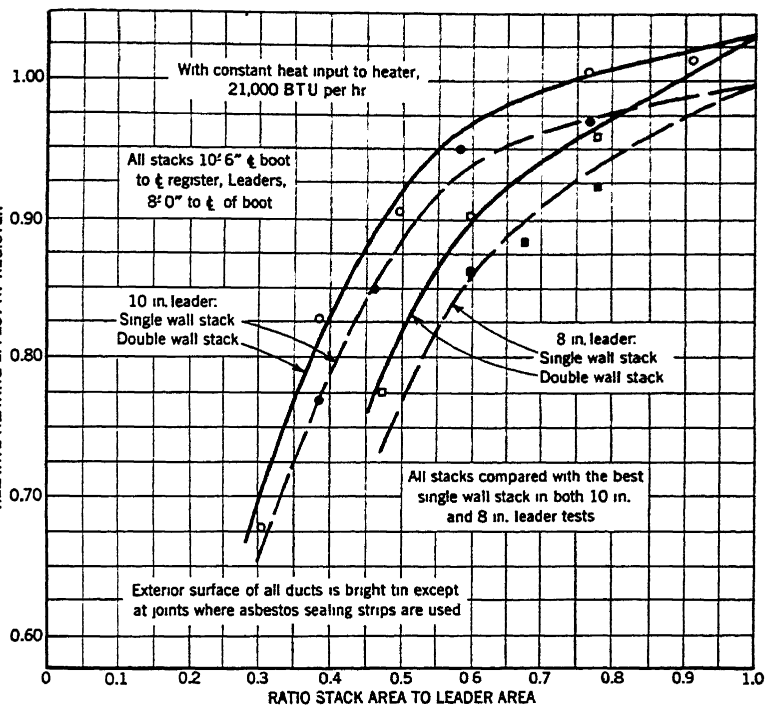

FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

sidewall locations for warm air registers in gravity circulating systems are not recommended on account of the tendency for stratification of the air in the room, resulting in high temperatures at the ceiling.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, in excess of the total area of warm-air pipes, and at all points where

the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return is placed to effectively receive the cold air returning by way of the stairs.

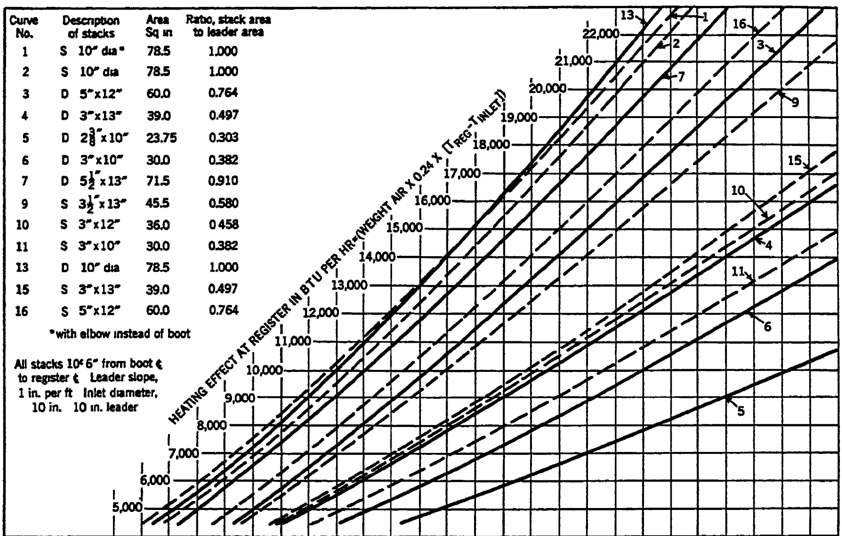

FIG. 4. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made oversize. This precaution is particularly important when long ducts and short ducts are used in the same system. The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces which are closed off or badly exposed. Metal linings are

advisable in such ducts. It is important that these ducts be free from unnecessary friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

Furnace Return Connection

Circulation of the air is accelerated if the return connection to the furnace is through a round inclined pipe connected to two 45 deg elbows rather than through a vertical pipe connected to two 90 deg elbows. The top of the return shoe should enter the casing below the level of the grate in the furnace. In order to accomplish this the shoe must be wide as is indicated in Fig. 6, No. 1 arrangement.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

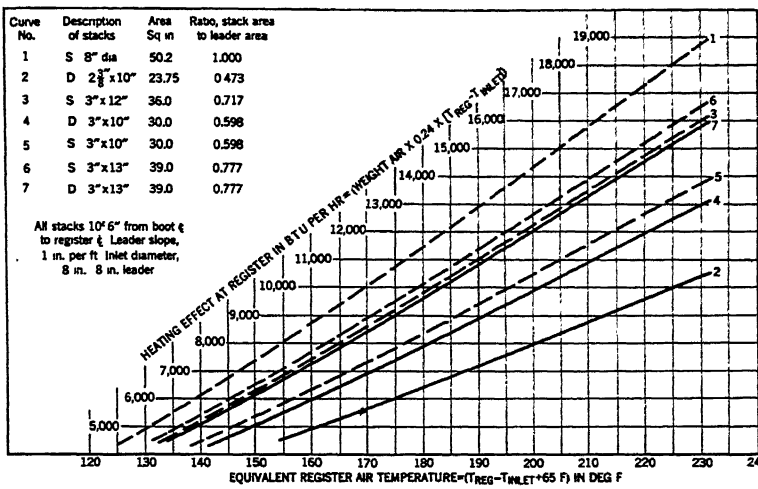

FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.
2. Friction and turbulence in elaborate return duct systems retard the flow of air and may seriously reduce furnace efficiency, and lessen the advantages of such a design.
3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is obtained by finding the sum of the leader pipe areas as already designated.

[2]Investigation of Warm Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz, and V. S. Day (University of Illinois, *Engineering Experiment Station Bulletin* No. 189).

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken at 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by a set of typical furnace performance curves (Fig. 7) for a cast-iron, circular radiator furnace with a 23 in. diameter grate and 50 in. diameter casing. The third factor is efficiency, which is a function of the combustion rate, and varies with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

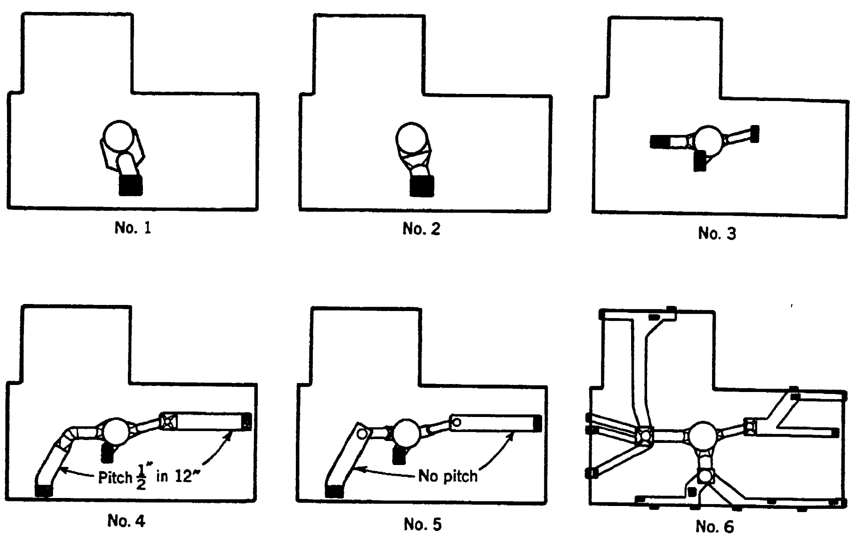

FIG. 6. ARRANGEMENT OF COLD AIR RETURNS FOR SIX INSTALLATIONS

It may be noted from Fig. 7 that for this particular furnace a register temperature of 175 F was accompanied by a combustion rate of approximately 7.5 lb per sq ft per hr, a capacity at the bonnet of 152,000 Btu per hr and a furnace efficiency of 58 per cent. Under these conditions the capacity at the bonnet per square foot of grate was equivalent to a value of 52,800 Btu per hr and per square inch of grate was equivalent to 367 Btu per hr. If it is desired to use these curves to select a furnace to deliver air at 175 F register temperature in a house where the total heat loss is H Btu per hour and the loss between the furnace and the registers is 0.25 H Btu per hour, the area of the grate in square inches will be $\frac{1.25\,H}{367} = 0.0034\,H$.

If, on the other hand, it is desired to select a furnace to deliver air at 160 F register temperature, the combustion rate is 5.5 lb and the efficiency

of the furnace is 62 per cent. Under this condition the capacity at the furnace bonnet per square foot of grate is 43,200 Btu per hr and per square inch of grate is 300 Btu per hr, the required area of the grate in square inches in this case will be $\frac{1.25\,H}{300} = 0.0042\,H$. It should be noted that a larger grate area is required if the furnace is to deliver air at a lower register temperature.

The typical performance curves shown in Fig. 7 are not applicable to

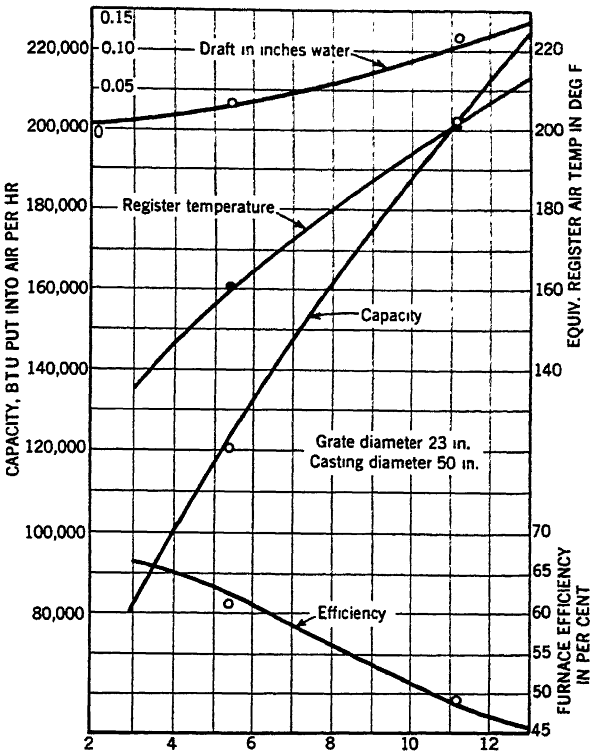

FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM-AIR FURNACE AND INSTALLATION IN A THREE-STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

all furnaces and hence for ordinary design purposes the values recommended in the Standard Code[3] should be used. The equation for a furnace having a ratio of heating surface to grate area of 20 to 1 is equal to:

$$H = \frac{G \times p \times f \times E_1 \times E_2 \times 0.866}{144} \tag{7}$$

[3]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the ninth edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 50 W. Broad St., Columbus, Ohio.

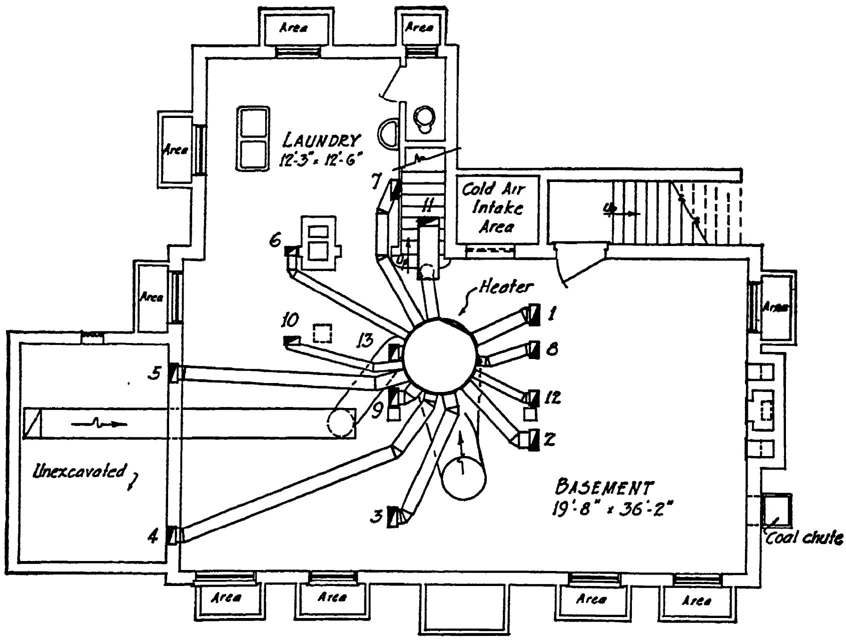

FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

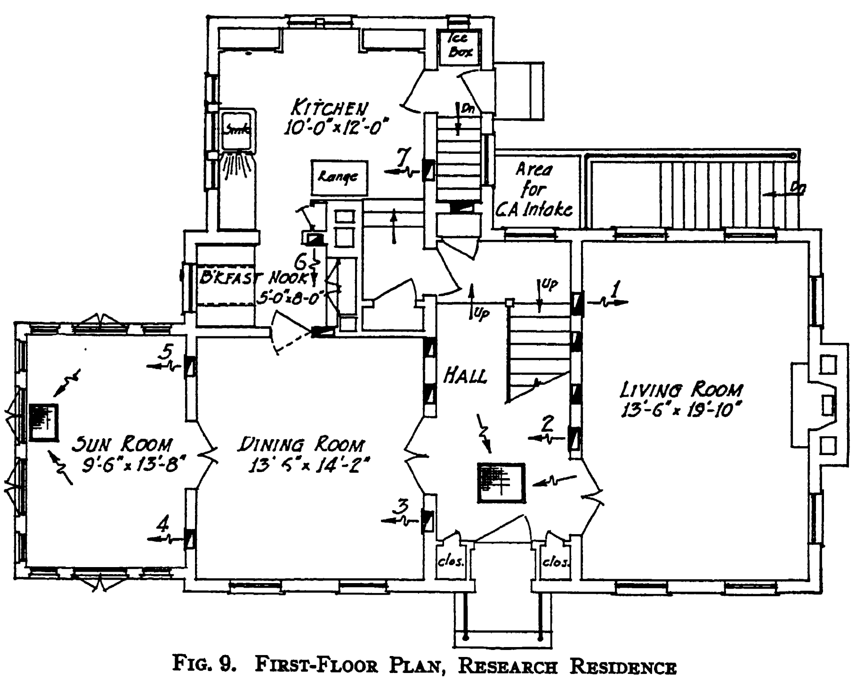

FIG. 9. FIRST-FLOOR PLAN, RESEARCH RESIDENCE

where

G = grate area, square inch.

p = combustion rate, pound coal per square foot of grate per hour.

f = heating value of the coal, Btu per pound.

E_1 = efficiency at bonnet, ratio of heat delivered at bonnet to heat developed in furnace.

E_2 = efficiency of duct transmission, ratio of heat delivered at register to heat delivered at bonnet.

0.866 = factor of safety to allow for contingencies under service conditions such as accumulations of soot and ashes, ineffective firing methods, etc.

H = total heat loss from structure.

An addition of 2 per cent of the furnace capacity is proposed for each unit that that ratio of heating surface to grate area exceeds 20. This addition is based on tests[4] conducted at the University of Illinois on seven types of furnaces having varying ratios of heating surface to grate area. This correction does not, however, apply to values of the ratio less than 15 nor greater than 30.

By transposing the terms in Equation 7 and adding the correction term for ratios of heating surface to grate area other than 20 to 1, the following equation is obtained:

$$G = \frac{144 \times H}{p \times f \times E_1 \times E_2 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{8}$$

in which R = ratio of heating surface to grate area.

In the case of the Standard Code[5] the numerical values used in Equation 8 were based on those determined from the tests conducted on the different types of furnaces.

$$G = \frac{144 \times H}{7.5 \times 12{,}790 \times 0.55 \times 0.75 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{9}$$

$$G = 0.004205 \frac{H}{[1 + 0.02\ (R\text{-}20)]} \tag{10}$$

As used in these calculations, H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the outside air, if any, at intake. This outside air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have a outside air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 25 per cent loss between furnace and registers.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

[4]University of Illinois *Engineering Experiment Station Bulletin* No. 246, by A. C. Willard, A. P. Kratz, and S. Konzo, Chapter X, pp. 126-146.

[5]Loc. Cit. Note 3.

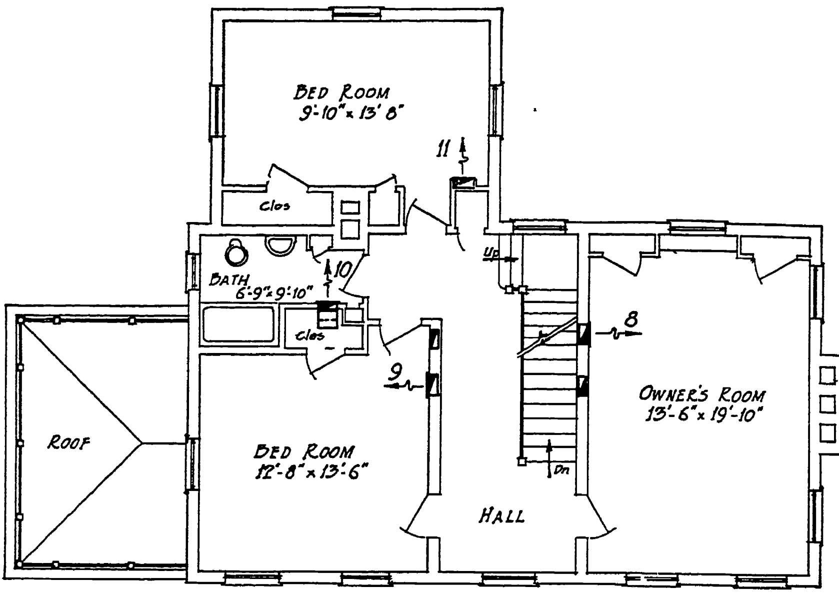

FIG. 10. SECOND-FLOOR PLAN, RESEARCH RESIDENCE

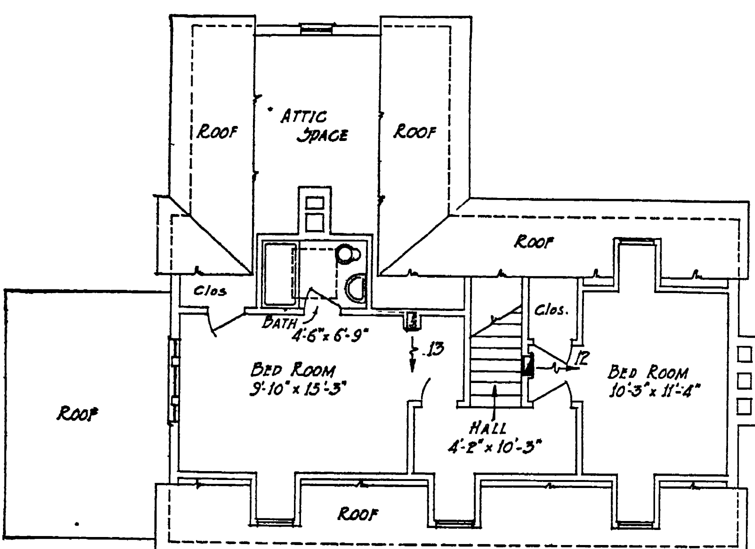

FIG. 11. THIRD-FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating and Air Conditioning Association* erected at the University of Illinois[6].

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in. leader area. See summary, Table 1; also example under Standard Code[7], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 in. × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See summary Table 1; also example under Standard Code[7], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 11.4, say 12 in.

Stack area = 0.7 × 90 = 63 sq in. = say 5 in. × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 in. × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code[7] Method. See Art. 3, Sec. 1, 2, 3)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12} + \frac{405}{60} + \frac{2405}{800}\right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12} + \frac{394}{60} + \frac{2275}{800}\right) 6 = 90 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate area = 0.0042 × 132,370 = 556 sq in.
Use 27 in. diameter grate. (Equation 10).

If provision should be made for certain outside air circulation, then increase the building heat loss by, say 25 per cent and obtain by Equation 10 a 30 in. grate.

Experiments at the University of Illinois[8] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan, with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft, and the ability of the fan to deliver an adequate amount of air.*

[6]Plans used with permission. Bathroom on third floor not heated.

[7]Loc. Cit. Note 3.

[8]University of Illinois, *Engineering Experiment Station Bulletin* No. 120, p. 129.

TABLE 1. SUMMARY OF DATA APPLIED TO WARM AIR RESEARCH RESIDENCE

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq In.	Stack Area Sq In. 0.7 × LA	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		= 0.009H				
Living	17250	155	—	14		14 × 16
Dining	6810	61	—	9		8 × 12
Breakfast	2300	21	—	8		8 × 10
Kitchen	9210	83	—	11 or 12		12 × 14
Sun	25710	230	—	Two 12		Two 12 × 14
Hall and stair	12570	113	—	12		12 × 14
Second Floor		= 0.006H				
Owner's	15030	90	63	11 or 12	5 × 12	12 × 14
S. W. Bed	9800	59	41	9	3½ × 12	8 × 12
Bath	2450	15	10	8	3 × 10	8 × 10
N. Bed	14800	89	62	11 or 12	5 × 12	12 × 14
Third Floor		= 0.005H				
E. Bed	8220	41	29	8	3 × 10	8 × 10
W. Bed	8220	41	29	8	3 × 10	8 × 10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[9].

[9]University of Illinois, *Engineering Experiment Station Bulletin* No. 141, p. 79, and No. 246.

PROBLEMS IN PRACTICE

1 ● What may prohibit the use of a gravity warm air system in a large house having several exposed wings?

In a gravity warm air system, excessive vertical distances above the furnace cause little trouble in the design of the wall stacks, but excessive horizontal distances from the furnace should be carefully considered in the design of the leaders. To work effectively, a gravity warm air system should be balanced and leaders over 12 ft in length should be avoided if possible. Long leaders, if used, must be of ample size, well pitched, and well insulated. Large houses having exposed wings may require leaders much longer than 12 ft; infiltration may create severe back-drafts in the exposed wings; and the basement ceiling height may not be sufficient to allow the leaders to have a pitch of more than one inch per foot. These conditions may make the exposed wings very difficult to heat with a gravity system because of its low air head differentials.

2 ● A first story dining room has a calculated heat loss of 12,000 Btu per hour.

a. What size leader pipe should be used for 175 F register air temperature?

b. What size register?

a. Leader area $= \dfrac{12{,}000}{111} = 108.1$ sq in. Use leader with diameter of 12 in.

b. Register gross area $= \dfrac{108}{0.7} = 154$ sq in. Use 12 in. by 14 in. register.

3 ● A third-story bedroom has a calculated heat loss of 12,000 Btu per hour.
a. What size leader pipe should be used for a 175 F register air temperature?
b. What size stack?
c. What size register?

a. Leader area $= \dfrac{12{,}000}{200} = 60$ sq in. Use leader with diameter of 9 in.

b. Stack area $= 0.7 \times 60 = 42$ sq in. Use stack 3½ in. by 12 in.

c. Register gross area $= \dfrac{60}{0.7} = 85.7$ sq in. Use register 8 in. by 12 in.

4 ● The calculated heat loss of a house is 130,000 Btu per hour. Find the grate area required for the furnace under the following conditions:

Heating value of coal = 12,790 Btu per pound.
Furnace efficiency = 55 per cent.
Combustion rate = 7.5 lb per sq ft per hr.
Ratio of heating surface to grate area of furnace = 20 to 1.
Register temperature = 175 F.
Loss between furnace and registers = 25 per cent.

See Equations 9 and 10:
Grate area $= 0.004205 \times 130{,}000 = 547$ sq in.
Grate diameter = 26.3 in.
Use grate with diameter of 26 in.

5 ● If in Question 4 the conditions were the same except that the ratio of heating surface to grate area of furnace was 24 to 1, what size grate would be required for the furnace?

$$\text{Grate area} = \frac{0.004205 \times 130{,}000}{1 + 0.02\,(24\text{-}20)} = \frac{547}{1.08} = 506 \text{ sq in.}$$

Grate diameter = 25.4 in.
Select grate with diameter of 25 in.

6 ● Name the items involved in the design of a furnace heating system.

a. Heat loss from each room, Btu.
b. Area and dimensions of warm-air pipes in basement, inches.
c. Area and dimensions of vertical pipes, inches.
d. Free and gross area and dimensions of warm-air registers, inches.
e. Area and dimensions of recirculating or outside air ducts, inches.
f. Free and gross area and dimensions of recirculating registers, inches.
g. Size of furnace necessary to supply the warm air to overcome the heat loss.
h. Area and dimension of chimney and smoke pipe, inches.

7 ● Discuss the design features of recirculating ducts.

a. Their area should be equal to or greater than that of the supply ducts.
b. They should be streamlined, and have a minimum number of turns.
c. All runs should be as short as possible.
d. Account should be taken of all cold walls and window areas in determining sizes and positions of return air inlets.

e. The return line should be pitched downward toward the furnace. It should be designed to minimize friction.

f. The top of the shoe or boot should never be above the grate level.

8 ● Discuss the use of a booster fan. What effect has a booster fan at low operating temperatures? At high ones?

A booster fan is useful in accelerating the air flow past the surface of a low temperature furnace, where only a small weight differential in the air is created, and in unbalancing a gravity system so flow is established. The first use involves the entire plant, and increases efficiency about 10 per cent with low temperature operation; the second involves only the leaders in which air flow is accelerated. At high operating temperatures the difference in weight between warm outgoing air and cool incoming air is great enough to make a booster unnecessary with ordinary gravity systems.

9 ● Is it desirable to use high side wall locations for warm air registers in gravity circulating systems?

High side wall locations are not recommended on account of the tendency for stratification of the air in the room resulting in high temperatures at the ceiling.

Chapter 20

MECHANICAL WARM AIR FURNACE SYSTEMS

Furnaces, Fans and Motors, Sound Control, Air Washers and Filters, Air Distribution Design, Automatic Controls, Design of Heating System, Selecting the Furnace, Selecting the Fan, Heavy Duty Fan Furnaces, Humidification, Cooling Methods, Cooling System Design

MECHANICAL warm air or fan furnace heating systems[1], which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 19. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by air washers or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the co-ordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

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[1]See University of Illinois, *Engineering Experiment Station Bulletin* No. 266 by A. P. Kratz and S. Konzo for details of tests conducted in Warm Air Research Residence.

FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel.
   b. *Anthracite or coke*—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Furnace Casings

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should be lined with black iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2 in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be

used, many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. The method of making these baffles for furnaces with top horse-shoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows the usual method of baffling square furnace casings for fan-furnace work.

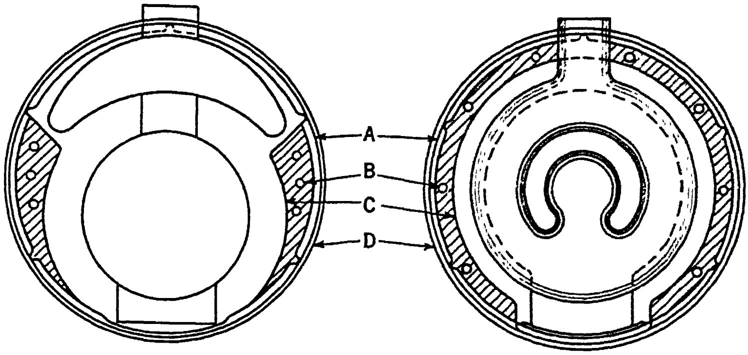

FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK

A. Liner, 1 in. from casing. *B*. Hole to vent baffle.
C. Baffle, closed top and bottom. *D*. Outer casing.

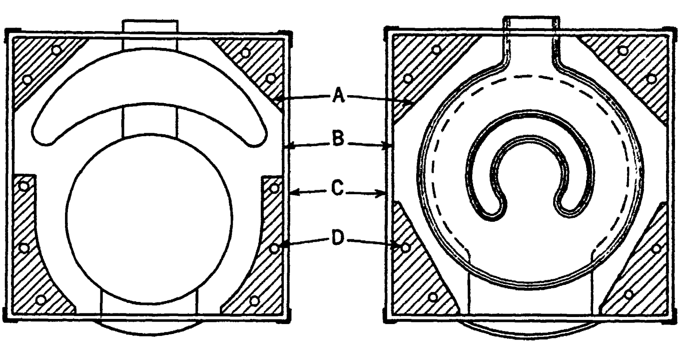

FIG. 2. METHOD OF BAFFLING SQUARE FURNACE CASING FOR FAN FURNACE WORK

A. Baffle, closed top and bottom. *B*. Liner, 1 in. from casing. *C*. Outer casing. *D*. Hole to vent baffle.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet

and thus provides a larger plenum chamber. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Low tip speed is desirable for the elimination of air noise, especially where forward curved blades are used. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys are desirable to provide a factor of safety and to allow for increased air circulation. For additional information on fans and motors, see Chapters 27 and 38.

SOUND CONTROL

Special attention should be given to the problem of noise elimination. The fan housing should not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. See also Chapter 30.

AIR WASHERS AND FILTERS

Washers for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Washers are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Washers used in connection with commercial or heavy duty plants should be a regulation type of commercial washer.

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. See also Chapter 26.

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. Due to the economic considerations involved, it is common practice to locate the supply openings on the inside walls of a residence and the return openings nearest the greatest outside exposure. Many designers prefer, however, to locate the supply registers so that the warm air from the registers *blankets* a cold wall, and mixes with the cold air dropping off from the

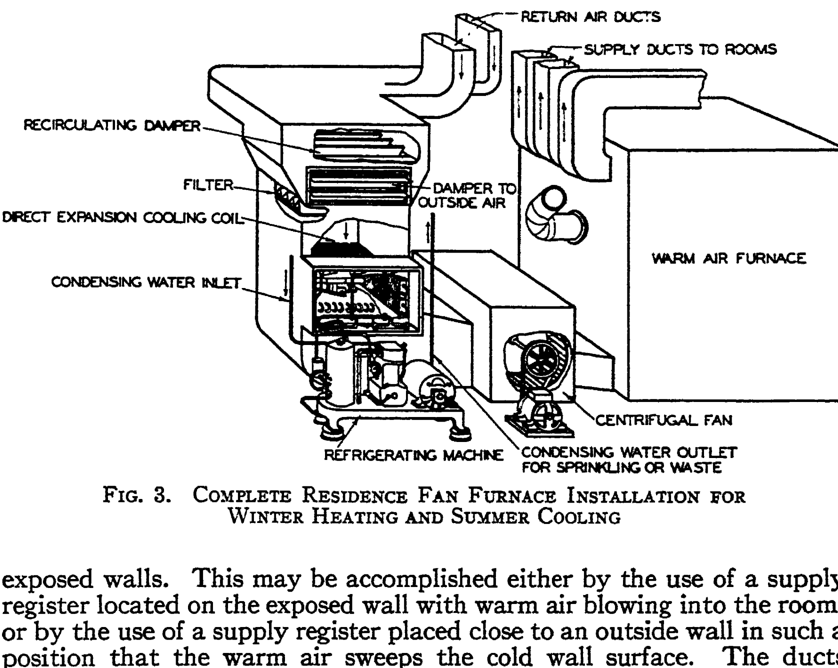

FIG. 3. COMPLETE RESIDENCE FAN FURNACE INSTALLATION FOR WINTER HEATING AND SUMMER COOLING

exposed walls. This may be accomplished either by the use of a supply register located on the exposed wall with warm air blowing into the room, or by the use of a supply register placed close to an outside wall in such a position that the warm air sweeps the cold wall surface. The ducts leading to supply registers which are located on the exposed walls should be adequately insulated to reduce the heat loss from the ducts.

Register and Grille Openings

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Tests conducted in the Warm Air Research Residence[2] have indicated that excellent results are obtainable with the use of a deflecting-diffuser type of baseboard register which throws the air downward toward the floor and diffuses the air at the same time. Unless registers located in the baseboard are well proportioned

[2]Loc. Cit. Note 1.

nd designed to harmonize with the trim, they may be unsightly. Better ir distribution for cooling is obtained when high side wall registers are sed, and this same location is satisfactory for heating when the openings re installed at least 7 ft above the floor line, providing the air velocity hrough the registers is greater than 600 fpm. Registers which are ocated in side walls above the baseboard or in the ceiling should be of an ffective air-diffusing type. All registers should be equipped with ampers, and should be sealed against leakage around the borders or nargins.

Velocities through registers may be reduced by the use of registers arger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated n Fig. 4. Merely to use a larger register may not result in materially educed velocities unless diffusers are used.

ampers

Suitable dampers are essential to any trunk or individual duct system, s it is virtually impossible to so lay out a system that it will be absolutely n balance without the use of dampers. Special care must be used in the

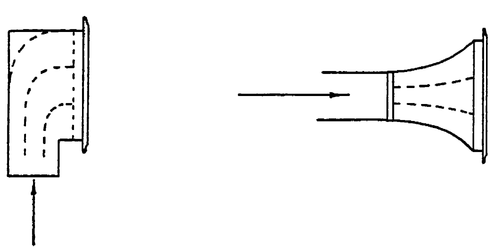

FIG. 4. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

esign of any system to avoid turbulence and to minimize resistance. harp elbows, angles, and offsets should be avoided. See Figs. 1 and 2, Chapter 29.

Three types of dampers are commonly used in trunk and individual uct systems. *Volume dampers* are used to completely cut off or reduce ne flow through pipes. (See *A* and *B*, Fig. 5.) *Splitter dampers* are used here a branch is taken off from a main trunk. (See *C*, Fig. 5.) *Squeeze ampers* are used for adjusting the volume of air flow and resistance nrough a given duct. (See *D*, Fig. 5.) It is essential that a damper be rovided for each main or duct branch. A positive locking device should e used with each type of damper.

ucts

The ducts may be either round or rectangular. Rectangular ducts nould be as nearly square as possible; the width should not be greater nan four times the breadth. The radii of elbows should be not less than ne and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in the system can be largely eliminated through proper care in the planning and installation of the control system.[3] The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.
3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.
4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To prevent cold air delivery when heat supply is insufficient.
6. To avoid heat loss through the chimney by keeping stack temperatures low.

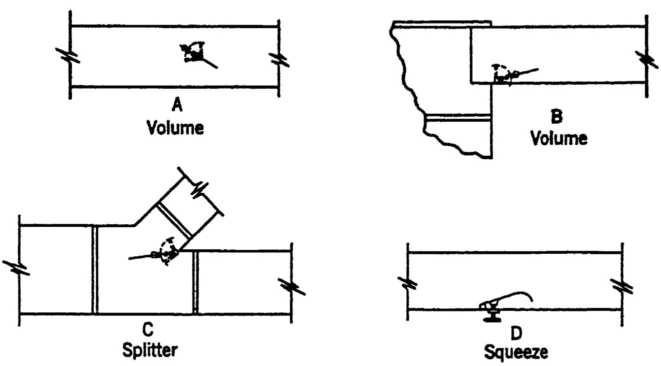

FIG. 5. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

7. To provide quick response to the thermostat, with protection against overrun.
8. To provide for humidity control.
9. To provide a means of summer control of cooling.
10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.

2. A *furnacestat* located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the furnacestat is in the main duct near the frame opening from the bonnet.

[3]Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 225 F.

4. On oil and gas burner installations, a control is usually included which will shut down the system if the fire goes out or if there is a failure of the ignition system.

5. A *humidistat* to regulate the moisture supplied to the rooms.

6. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying.

METHOD OF DESIGNING FORCED-AIR HEATING SYSTEMS

1. Determine heat loss from each room in Btu per hour. (See Chapter 7).

2. Locate warm air registers and return registers on plans of house, beginning with the upper story rooms.

3. Sketch in duct layout to connect all registers and grilles with the central unit.

4. Determine equivalent length of duct for each register, allowing 10 diameters of straight pipe as equivalent to each 90 deg elbow having an inner radius not less than the diameter of the round pipe or the depth of the rectangular pipe.

5. Select a value for temperature of the air at the furnace bonnet. It is customary to use some value lying between 150 to 165 F. Use lower value if larger number of air recirculations is desired. It is recommended that the number of air recirculations should be in excess of 5 per hour.

6. Determine approximate value of temperature reduction in each duct caused by heat loss from the ducts. A value of from 0.3 to 0.6 F per foot of duct has been obtained from tests conducted in the Research Residence installation for uninsulated duct lengths up to approximately 60 ft.

7. Subtract this temperature reduction from the assumed bonnet air temperature to obtain an approximate value of the register air temperature for each register.

8. Determine the required air volume for each room from the following equation, or from the values listed in Table 1:

$$Q = \frac{H}{60 \times 0.24 \times d\,(t_r - 65)} \qquad (1)$$

where

Q = required air volume, cubic feet per minute.
H = heat loss of room, Btu per hour.
d = density of air at register temperature, pounds per cubic foot.
t_r = register temperature, degrees Fahrenheit.
0.24 = specific heat of air.
65 = return air temperature.

For any given register temperature the solution of this equation simplifies to the following form:

$$Q = H \times \text{Factor} \qquad (2)$$

in which the values of the Factor may be obtained from Table 1.

9. Determine register size from the air volume delivered to each room by the following formula:

$$\text{Free area of register, square feet} = \frac{Q}{V} \qquad (3)$$

$$\text{Gross area of register, square feet} = \frac{\text{Free Area}}{R} \qquad (4)$$

where

Q = required air volume, cubic feet per minute.
V = velocity at register face, feet per minute.
R = ratio of free area to gross area of register.

Table 1. Factors Corresponding to Register Temperature for Equation 2

Register Temperature	Factor
110	0.02210
120	0.01840
130	0.01585
140	0.01397
150	0.01253
160	0.01140
170	0.01049

Allowable register velocities to be used in Equation 3 are approximately as follows:

Baseboard, non-deflecting type, maximum = 300 fpm.
Baseboard, deflecting toward floor, maximum = 500 fpm.
Baseboard, deflecting and diffusing = up to 800 fpm.
High sidewall = not less than 600 fpm.

10. Duct systems for forced-air installations may consist of either trunk systems or individual duct systems.

Trunk Systems. Determine duct sizes and friction losses as outlined in Chapter 20, except that for residence applications the velocities in the main duct and in the various parts of the system should approximate the values recommended in Table 2.

Individual Duct Systems. An individual duct system is one having separate ducts extending from the heating unit to each register. In designing such a system select first the duct having the greatest equivalent length. Select a reasonable velocity using Table 2 as a guide. From friction chart on page 566 determine unit friction loss per 100 ft of run, and from this the total friction loss in the duct selected. If this total friction loss exceeds a reasonable value a lower velocity should be used.

The remaining ducts are proportioned so that the total pressure in each duct is the same as that calculated for the longest duct. The added resistance necessary in the shorter ducts is accomplished by increasing the velocity in these ducts. No duct should be less than 6 in. in diameter, nor should the velocity in any duct exceed approximately 1200 fpm. The final adjustment in a duct system may be made by employing dampers.

Instead of proportioning the ducts as outlined in the preceding paragraph it is more usual in practice to proportion all the ducts so that they have the same velocity as that used in the longest duct and to balance the system by employing dampers in the shorter ducts.

Return duct systems are designed making use of the same principles as those used in the design of supply duct systems. In this case the design may be based on the volume of air corresponding to the density of air existing in the return ducts, or in order to provide a factor for air leakage, it may be based on the same volume as used for the supply ducts.

Table 2. Recommended Velocities Through Ducts and Registers

Description	Low Velocity System (fpm)	Medium Velocity System (fpm)	High Velocity System (fpm)
Main ducts	500	750	1000
Branch ducts	450	600	750
Wall stacks	350	500	600
Baseboard registers (max.)	300	350	400
Wall registers above 5 ft (min.)	500	550	600

11. Determine frictional resistance in:

a. Supply side of system as outlined in item 10.
b. Return side of system as outlined in item 10.
c. Furnace units, casing or hood, which is usually considered as equivalent to 0.03 to 0.10 in. of water.
d. Accessories such as washers or air filters, from manufacturer's data.
e. Inlet and outlet registers and grilles, from manufacturer's data.
f. Other accessory equipment such as cooling coils, from manufacturer's data.

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the items listed in the preceding discussion. In practice it is recommended that liberal allowances should be made so that the fan will be capable of delivering air against pressures that may not have been foreseen during the design of the duct system.

12. Select a furnace capable of delivering heat at the register outlets equal to the total heat loss of the structure to be heated.

The following formula may be used for coal burning furnaces:

$$G = \frac{H}{f \times p \times E_1 \times E_2 \,[1 + 0.02\,(R - 20)]} \tag{5}$$

where

G = required grate area, square feet.
H = total heat loss from building, Btu per hour.
f = calorific value of coal, Btu per pound.
p = combustion rate in pounds of fuel per square foot of grate per hour.
E_1 = furnace efficiency based on heat available at bonnet.
E_2 = efficiency of transmission based on ratio of heat delivered at register to heat available at bonnet.
R = ratio of heating surface to grate area.

In practice it is customary to use the following constants:

f = 12,000 (for specific values, see Table 1, Chapter 9).
p = 7.5 lb.
E_1 = 0.65 lower efficiency must be used with highly volatile solid fuel.
E_2 = 0.85.

The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

13. Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick. Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

14. Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturers' Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

15. The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula:

$$R = \frac{H}{0.9} \tag{6}$$

where

H = total heat loss from building, Btu per hour.
R = official *A. G. A.* output rating of the furnace, Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.59\ H \tag{7}$$

where

I = Btu per hour input.

The factor 1.59 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

16. Specify location and type of all dampers in both supply air and return air sides of system. Specify controls including location of all thermostats. Arrange for proper control of humidifying equipment.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. A few possible arrangements are shown in Figs. 6 to 9 inclusive.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 21. Ducts are designed by the method outlined in Chapter 29.

HUMIDIFICATION

Mechanical warm air systems offer a means of proportioning and distributing moisture-bearing air; consequently, during the winter months humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the

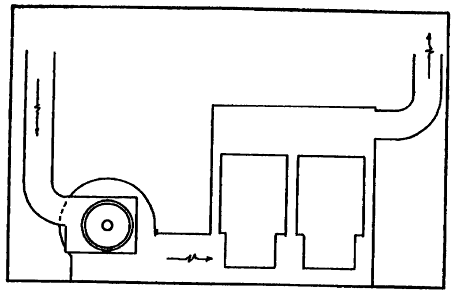

FIG. 6. HEATER ARRANGED FOR COMPLETE RECIRCULATION OF AIR

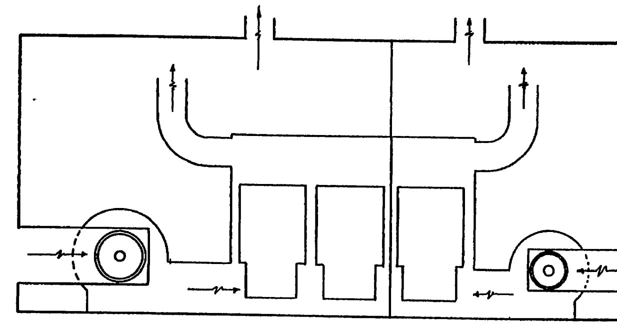

FIG. 7. TWO BATTERIES OF HEATERS AND FANS FOR INDEPENDENT SERVICE USING OUTSIDE AIR AND EXHAUSTING TO ATMOSPHERE

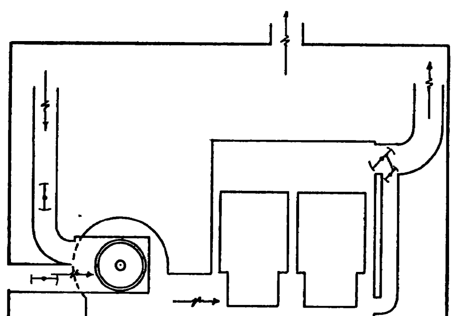

FIG. 8. HEATER ARRANGED FOR PARTIAL RECIRCULATION, ALSO SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR CHAMBERS, AND PARTIAL EXHAUST TO ATMOSPHERE

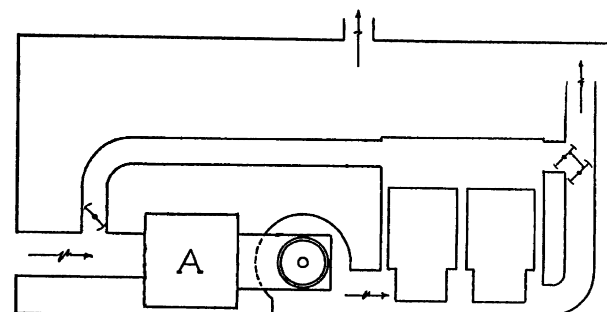

FIG. 9. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

limits of the generally accepted standards. See Chapters 3 and 25 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface.

Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 25.)

Residence Requirements

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin No.* 230[4], as follows:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 65 deg[5] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69.5 F with relative humidity of 40 per cent is the most practical for the attainment of 65-deg effective temperature.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5. The problems of humidity requirements and limitations cannot be separated from considerations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of gravity warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

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[4]See Humidification for Residences, by A. P. Kratz (*University of Illinois, Bulletin No.* 230).

[5]66 deg is the optimum winter effective temperature recommended by the A.S.H.V.E. Committee on Ventilation Standards.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. See also Chapters 22 and 24.

Results at Research Residence

The following conclusions may be drawn from the studies thus far completed in the Research Residence, subject to the limitations of the conditions under which the tests were run[6].

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10 year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the frictional resistance of the coil to flow of air must be given careful consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

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[6]Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 95); Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 167).

METHOD OF DESIGNING COOLING SYSTEM

The general procedure for the design of a cooling system in a forced-air installation is as follows:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 5 and 8). Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests[7] a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry bulb and 45 per cent relative humidity was found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 22).

5. Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the heat to be removed by the cooling unit, in the form of sensible heat and latent heat.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter under the heading of Method of Designing Forced-Air Heating Systems.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. See Chapter 22 on section Surface Type Dehumidifier.

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

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[7]Loc. Cit. Note 6.

PROBLEMS IN PRACTICE

1 ● A residence furnace, having a ratio of heating surface to grate area equal to 20 to 1, is to be selected to heat a house which has a computed load of 225,000 Btu per hour. If coal having a calorific value of 12,000 Btu per pound is to be burned, if the furnace will burn 7.5 lb of coal per square foot of grate per hour, and if the furnace efficiency is 65 per cent, determine the square feet of grate area necessary in the furnace to be selected.

Substituting in Equation 5:

$$G = \frac{225{,}000}{12{,}000 \times 7.5 \times 0.65 \times 0.85} = 4.53 \text{ sq ft of grate area.}$$

A furnace having at least 4.5 sq ft of grate area should therefore be selected.

2 ● Why should secondary surface be designed for easy cleaning?

If the combustion is not perfect, soot is formed immediately above the fire and is apt to form a deposit on the secondary surface from which it should be removed. If the secondary surface is so designed that there are horizontal passages, fine gray ash will settle out in these to form an insulation between the hot gases of combustion and the metal of the furnace; consequently, these should be readily cleaned. If the passages are vertical they are largely self-cleaning of ash, but provision should be made for easy and thorough cleaning of the collection chamber below them.

3 ● Why is baffling inside the casing necessary on fan systems?

Because the movement of air is independent of its temperature, air must be guided by baffles of one form or another to bring it in contact with the hot surfaces so it will not pass through the casing unheated. On the other hand, if the air is held against a hot surface too long it might become overheated, for the average register temperature on a fan system should not exceed 120 F.

4 ● What practical points should be observed in designing a fan system in order to eliminate noise?

a. Use a large fan so it can be run at slow speed.

b. Set the fan and motor on a solid foundation.

c. Insulate the fan and motor from the foundation with rubber, cork, or other springy material according to the principles given in Chapter 30, provided, of course, that such insulation is of value.

d. See that the air velocity is not too high in the ducts. Properly designed splitters in the elbows will avoid high velocities at the turns in cases where the velocity through the ducts themselves is not too high.

e. Use canvas connections between the ducts and any running equipment.

f. Be sure the ducts have a relatively smooth interior and are rigid.

5 ● Why do furnaces designed to burn bituminous coal, oil, or gas require larger combustion spaces than those designed for anthracite?

Anthracite burns largely as fixed carbon whereas gas and oil burn as gases, and as much as 50 per cent of bituminous coal burns as a gas. Ample space must be provided for the intimate mixture of these gases with the oxygen of the air to secure proper combustion.

6 ● A furnace has the following dimensions: Grate diameter, 24 in.; casing diameter for gravity air flow, 56 in.; combustion chamber diameter, 30 in. What is the unobstructed area required for passage of air across the heating surface when a motor-driven blower, operating at an outlet velocity of 1200 fpm, delivers 1600 cfm into the casing near its bottom?

For residence applications using small blowers, an air outlet velocity of about one third of the blower outlet velocity is considered good practice.

$$\text{Air-pass velocity} = \frac{1200}{3} = 400 \text{ fpm.}$$

$$\text{Air-pass area} = \frac{1600}{4} = 4 \text{ sq ft} = 576 \text{ sq in.}$$

7 ● In Question 6 what would be the gap between the chamber and the baffle when the chamber is centered in the casing?

Area of combustion chamber (30-in. diam)	706.9 sq in.
Area of air pass	576.0 sq in.
Total area	1282.9 sq in.

The diameter of a circle with an area of 1282.9 sq in. is 40.4 in. One half of the difference between the diameters is the amount of gap.

$$\text{Gap} = \frac{40.4 - 30.0}{2} = 5.2 \text{ in.} = \text{approximately } 5\tfrac{1}{4} \text{ in.}$$

Chapter 21

CENTRAL SYSTEMS FOR HEATING AND HUMIDIFYING

Types of Systems, Blow-Through, Draw-Through, Heating Units, Design, Temperatures, Weight of Air to be Circulated, Temperature Loss in Ducts, Heat Supplied Heating Units and Washer, Grate Area, Boiler Selection, Weight of Condensate, Static Pressure, Fans and Control

A FAN system of heating depends upon fans and blowers to distribute air through ducts from one centrally located plant. This chapter considers heating and humidifying systems of this type whereas similar systems arranged for cooling and dehumidifying are discussed in Chapter 22. A special type of central fan system, the mechanical warm air or fan furnace system, which is especially adapted to residences, churches, halls, and other small buildings, is covered in Chapter 20.

TYPES OF SYSTEMS

In the indirect type of central fan heating and air conditioning systems, steam is usually the medium by which heat is transferred from the boiler, or other source of heat, to the heating units. If the system is intended solely for heating, the air is passed over one or more stacks or batteries of heating units and then conveyed to the spaces for which it is intended through a system of ducts. In some cases, a predetermined amount of outside air is introduced for ventilating purposes, whereas in others the moisture content is controlled by passing the air through a washer or humidifier. If the apparatus is designed to control simultaneously the temperature, humidity, air motion, and distribution, it is known as an air conditioning system.

In the *split system*, the heating is accomplished by means of radiators or convectors, and the ventilating or air conditioning by means of the central fan apparatus. In the *combined system*, the entire operation of heating, ventilating, and air conditioning is handled by the central fan system.

A common arrangement of the central fan system of heating is illustrated by Fig. 1 and consists of a fan, a heating unit (heater) enclosed by a sheet metal casing connected with the suction side of the fan, a sheet metal casing connected to the heating unit casing run to the outside of the building and provided with an adjustable opening inside the building for recirculation of the air when desired, and a duct system attached to the fan outlet to convey and distribute the air to various parts of the building to be warmed by the apparatus. The fan is ordinarily motor-driven; there are, however, many cases when a direct-connected steam engine may be used to advantage. In this event the exhaust from the engine can be con-

nected to one or more sections of the heater, depending upon the condensation rate of the engine. The recirculation duct connected with the opening in the suction duct should be extended to a point as near the floor as possible.

When ventilation is not a requirement or is considered relatively unimportant, as in shop and factory heating, and the number of persons vitiating the air is small compared with the cubical contents of the building, or the process does not generate obnoxious gas or vapors, the air may be recirculated, sufficient outside air for ventilation being supplied by infiltra-

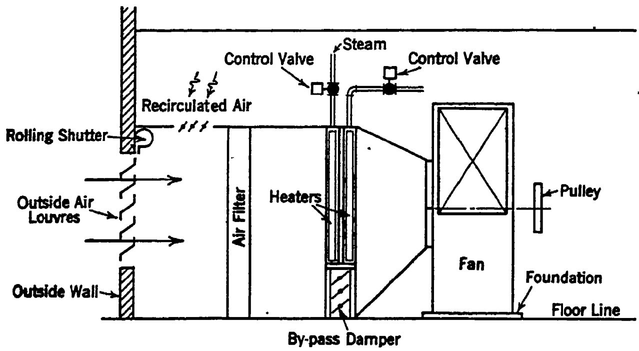

FIG. 1. ARRANGEMENT OF A CENTRAL FAN HEATING SYSTEM (DRAW-THROUGH)

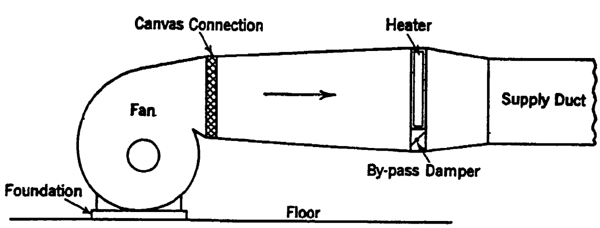

FIG. 2. ARRANGEMENT FOR HEATING UNIT (BLOW-THROUGH)

tion. The amount of heat to be supplied the heating unit in this case is the same as would be required for a direct radiation installation.

When ventilation is a requirement to be met, an arrangement similar to that shown by Fig. 1 may be employed. Since the amount of air necessary for heating is generally in excess of the amount required for ventilation, considerable fuel economy may be effected by recirculating a portion of the air. In this case only sufficient outside air is drawn into the system to meet the ventilation requirement and the remainder of the air, required for heating, is recirculated. This may be readily effected by an arrangement of ducts and dampers on the suction side of the fan as previously mentioned. If the outside air introduced is to be washed or conditioned the washer or humidifier and tempering coil may be added between the inlet for the recirculated air and the fresh air intake.

Blow-Through, Draw-Through

When the heating unit is located on the suction side of the fan, the system is known as *draw-through*. (See Fig. 1.) When the heating unit is located in the discharge from the fan, the system is known as *blow-through*. (See Fig. 2.) The draw-through combination is used for factory and toilet room installations because a more compact arrangement of the apparatus usually is possible. In addition, air leakage will be inward. The blow-through combination is used principally in schools and public buildings, and for all booster coil arrangements where different temperatures and independent temperature regulation are required for different heated spaces. In public building installations, the fan frequently blows the heated air into a plenum chamber from which the air ducts radiate to the various rooms of the building; this arrangement is sometimes called the *plenum* system.

HEATING UNITS

The heating units for central fan systems using steam as the heating medium may be classified as (1) tempering coils, (2) preheater coils, (3) reheater coils, (4) booster coils, and (5) water heaters, either open or closed. *Tempering coils* are used with ventilating and air conditioning systems for raising the temperature of the outside cold air to above freezing, or 32 F. They are not required for heating systems where all of the air is recirculated, since the temperature of the recirculated air will be above freezing. *Preheater coils* are used with air conditioning systems to raise the temperature of the air from that leaving the tempering coils to such a temperature that in passing through the water sprays of the washer (without water heater) the air will become partially saturated (adiabatically) having a moisture content corresponding to the required dew-point temperature. Preheater coils therefore supply heat as necessary to control the dew-point temperature. The *reheater coils* are used to raise the temperature of the air leaving the tempering coils (in the case of a heating or ventilating system) or the air leaving the washer (in the case of an air conditioning system) to that necessary to maintain the desired temperature in the rooms or spaces to be heated or conditioned, except where booster coils are used, in which case the reheater coils raise the air temperature to approximately room temperature, or slightly higher. *Booster coils* are installed in the duct branches to control the temperature of the air entering the rooms or spaces for which it is intended. *Water heaters* are used on an air conditioning system to control the dew-point temperature. They are used mainly for industrial work, seldom for comfort conditioning. They are not used where preheater coils are employed. The open type supplies steam directly to the spray water, while the closed type utilizes a heat interchanger by which the steam imparts its heat to the spray water. Where water heaters are required for comfort conditioning, the closed type is used.

The heating units for central fan systems in use at the present time consist either of pipe coils, finned tubes of steel, copper, brass or other metal, cast-iron sections with extended surfaces, or the cellular type. Steam is passed through these heating units and the air to be heated is passed over their exterior surfaces.

In selecting a heating unit for any particular service, the choice should be based on the desired requirements as follows:

1. Final temperature desired.
2. Loss in pressure (or friction) of air passing over the heating unit.
3. Air velocity over the heating unit.
4. Free area or face area of heating unit.
5. Ratio of heating surface to net free (or face) area.
6. Air volume required.
7. Number of rows of pipes, tubes, or sections.
8. Amount of heating surface.
9. Steam pressure drop through the heating unit.
10. Weight of heating unit.

Final Temperature Desired. The choice of a heating unit is largely influenced by the final temperature desired, when the entering air temperature and steam pressure available at the heating unit are specified. These data are obtainable from manufacturers' catalogs.

Loss in Air Pressure (or Friction). The allowable friction through the heating unit is one of the first factors to be determined in the selection of the apparatus. The velocities of air through various types of heating units will not necessarily be the same, but for any particular job the velocity through the heating unit should be a secondary consideration and the allowable friction or air pressure loss should be fixed approximately before proceeding with the selection of the heating unit. The loss in air pressure (or friction) through the heating unit should not exceed a predetermined maximum allowable amount for economical operation and for moderate size and first cost of installation.

In public building work, the maximum allowable friction through both tempering coils and reheater coils should never exceed ⅝ in. of water and it is advisable that the friction be kept considerably lower than this figure if possible. A tempering coil friction ranging from 0.10 to 0.20 in. of water is considered satisfactory. The air pressure loss for reheaters ordinarily ranges from 0.20 to 0.40 in. of water. In factory work, the maximum friction through the heater should never exceed 0.8 in. or 1 in. of water and it is advisable to figure the heaters at lower frictions if possible.

Velocity through Heating Unit. This velocity has generally been given in manufacturers' tables as being measured at 70 F and in most cases refers to the velocity through the net free area of the heating unit, or through the net space between the pipes, tubes or sections. Although most manufacturers give suitable velocities measured at 70 F, certain manufacturers show velocities measured at 65 F and others indicate velocities measured at the average air temperature through the heating unit. Many new heating units, however, specify net face areas with corresponding velocities instead of velocities through net free areas. In either case, manufacturers publish the corresponding friction or air-pressure loss in tables. The velocity through the net free area of the heating unit averages about 1000 fpm and that through the net face area about 500 fpm.

The *volume of air* to be heated in any particular case is determined after consideration of the ventilation requirements, heat losses, and quantity of air required for proper circulation, as explained in Chapters 3 and 7.

The number of *rows of pipes, tubes, or sections* or the amount of *heating surface* to be used may be selected from manufacturers' catalogs after the quantity of air handled and the heat load are known. Savings in operating expense or cost of installation should result from a proper selection of heater and by-pass areas. For example, instead of having the entire air quantity go through a one-row heating unit, it may be advantageous to use a two-row heating unit and a properly sized by-pass. Thus, when no heating is being done, a suitable by-pass damper may be opened to place a lighter load on the fan.

The *steam pressure drop through the heating unit* is also tabulated in manufacturers' data tables. The sizing of steam supply and return piping, allowing for drops through heating units, is explained in Chapter 16.

Weight of Heating Unit. In the design of a heating system, the weight limitations of heating units are determined by the location of the units. Obviously, if there is no loading limitation imposed, any type of heating unit may be selected. On the other hand if the heating unit is to be hung from the ceiling, it may be desirable to use the lightest unit which will accomplish the work required.

DESIGNING THE SYSTEM

The general procedure for the design of central fan systems is as follows:

1. Calculate the heat loss for each room or space to be heated.
2. Determine volume of outside air to be introduced.
3. Assume or calculate temperature of air leaving registers or supply outlets.
4. Calculate weight of air to be circulated.
5. Estimate temperature loss in duct system.
6. Calculate heat to be supplied the heating units and washer.
7. Select heating units and washer from manufacturers' data and performance curves.
8. Calculate total heat to be supplied.
9. Calculate grate area and select boiler.
10. Design duct system.
11. Calculate total static pressure of system.
12. Select fan, motor, and drive.

The heat losses (H) should be calculated in accordance with the procedure outlined in Chapter 7. If a positive pressure is maintained by the central fan system in the room or space to be ventilated or conditioned, there will ordinarily be very little infiltration of cold outside air through the cracks and crevices of the space. Consequently, the volume of air introduced into the space at the assumed or calculated outlet temperature need only be sufficient to provide for the transmission losses, plus about one-third of the infiltration losses. The exfiltration of heated or conditioned air through the cracks and crevices of the space should be provided for by making the usual allowance for the infiltration losses in arriving at the total heat loss of the space. The air required to make up for this exfiltration of heated or conditioned air will be brought in at the outside air intake and may be included as a part of the outside air neces-

sary for the ventilating requirements. The heat required to raise this air to the conditions maintained in the room must be provided by the tempering coils, preheater coils, and reheater coils. If a positive pressure is not maintained in the room or space to be conditioned, the normal infiltration of outside cold air will take place in this room, and the outlet temperature, together with the required air volume at this temperature, must be sufficient to provide for both infiltration and transmission losses.

Volume of Outside Air

The volume of outside air required for ventilation or air conditioning purposes may be determined from data in Chapter 3. In no case shall less than 10 cfm per person be introduced.

The heat required to warm the outside air introduced for ventilation purposes (H_o) may be determined by means of the following formula:

$$H_o = 0.24\ (t - t_o)\ M_o \tag{1}$$

where

0.24 = specific heat of air at constant pressure.
t = room temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
M_o = weight of outside air to be introduced per hour, in pounds = $60\ d_oQ_o$.
Q_o = volume of outside air to be introduced, cubic feet per minute.
d_o = density of air at t_o, pounds per cubic foot.

Example 1. A building in which the temperature to be maintained at 70 F requires 10,000 cfm. If the outside temperature is 20 F, how much heat will be required to warm the air introduced for ventilation purposes to the room temperature?

Solution. 10,000 $\times$ 60 = 600,000 cfh; d_o = 0.08273 (Table 1, Chapter 1); M_o = 0.08273 $\times$ 600,000 = 49,656 lb; t = 70 F; t_o = 20 F; H_o = 0.24 $\times$ (70 − 20) $\times$ 49,656 = 595,872 Btu per hour.

Temperature of Air Leaving Registers

If the system is to function only as a heating system, that is, entirely as a recirculating one, the temperature of the air leaving the register outlets must be assumed. For public buildings, these temperatures may range from 100 to 120 F, whereas for factories and industrial buildings the outlet or register temperature may be as high as 140 F. In no case should the outlet temperature exceed these values.

For ventilating or conditioning systems, the temperature of the air leaving the supply outlets may be estimated by means of the following formula:

$$t_y = \frac{H}{60\ d\ Q \times 0.24} + t \tag{2}$$

where

t_y = outlet temperature, degrees Fahrenheit.
H = heat loss of room or space to be conditioned, Btu per hour.
Q = *total* volume of air to be introduced at the temperature t, cubic feet per minute.
d = density of air, pounds per cubic foot.

If the outlet temperature (t_y) as determined from Equation 2 exceeds 120 F for public buildings, or 140 F for factories or industrial buildings,

these respective outlet temperatures should be used as factors in the following equation to determine the volume of air to be introduced into the room or space:

$$Q = \frac{H}{60\,d \times 0.24\,(t_y - t)} \tag{3}$$

Example 2. The heat loss of a certain auditorium to be conditioned is 100,000 Btu per hour. The ventilating requirements are 1,500 cfm and the room temperature 70 F. Determine the outlet temperature.

Solution. Substituting in Formula 2,

$$t_y = \frac{100{,}000}{60 \times 0.07492 \times 1500 \times 0.24} + 70 = 131.7 \text{ F}$$

Inasmuch as this temperature is excessive, it will be necessary to assume an outlet temperature, which will be taken as 120 F, and to calculate the amount of air to be introduced into the room at this temperature to provide for the heat loss. Substituting in Equation 3,

$$Q = \frac{100{,}000}{60 \times 0.07492 \times 0.24\,(120 - 70)} = 1850 \text{ cfm (at temperature } t)$$

Weight of Air to be Circulated

The total weight of air (M) to be introduced into the room or space to be heated or conditioned is given by the following formulae:

$$M = \frac{H}{0.24(t_y - t)} = 60\,dQ \tag{4}$$

$$M = M_o + M_r \tag{5}$$

$$M_o = 60\,d_oQ_o \tag{6}$$

where

d = density of air at temperature t, pounds per cubic foot.
d_o = density of air at temperature t_o, pounds per cubic foot.
Q_o = volume of outside air at temperature t_o, cubic feet per minute.
M_o = weight of outside air, pounds per hour.
M_r = weight of recirculated air, pounds per hour.

Example 3. Using the data of Example 2 and an outside temperature of 20 F, what will be the values of M, M_o and M_r?

Solution. $d = 0.07492$; $d_o = 0.08273$; $Q = 1850$; $Q_o = 1500$; $H = 100{,}000$.

$$M = \frac{100{,}000}{0.24 \times (120 - 70)} = 8{,}333 \text{ lb}$$

$$M_o = 0.08273 \times 60 \times 1500 = 7{,}448 \text{ lb}$$

$$M_r = M - M_o = 8{,}333 - 7{,}448 = 885 \text{ lb}$$

Temperature Loss in Ducts

The allowances (t_z) to be made for temperature drop through the duct system are as follows:

1. When the duct system is located in the enclosure to which the air is being delivered, as in a factory, it may be assumed that there is no loss between the reheater coil and the point or points of discharge into the enclosure.

2. For ducts in outside walls, basements, attics or other exposed places temperature drops should be determined in accordance with the procedure as outlined in Chapter 39.

3. For ducts run underground an allowance shall be made based on the assumption that the average ground temperature will be 55 F.

Heat Supplied Heating Units and Washer

The following cases may arise in practice:

A. The heating of the building is done entirely by means of a central fan system, all of the air being drawn from the outside.

B. Similar to *A*, except that all of the air is recirculated.

C. A portion of the air is recirculated, and the remainder is drawn in from the outside.

D. Air at the same temperature is to be delivered to all the rooms. A constant relative humidity is maintained in the building and all of the air circulated is drawn from outside the building. (Not applicable to the heating of various rooms where individual control of each room is desired.)

E. Outside air, return air, and by-pass air are used with the reheater located in by-pass air chamber.

F. Arrangement of apparatus where individual control of the temperature for each room is required in conjunction with air washer equipment to maintain a constant relative humidity in the rooms. The air washer is provided with a water heater for the spray water, capable of fully saturating the air. A section of preheater may be used for this purpose in place of the water heater. With this arrangement and with a uniform temperature of air entering the rooms, it is impossible to maintain the same room temperature throughout the building because the weight of air to be delivered to each room is determined and fixed by the ventilating requirements.

In analyzing these cases, the following symbols will be used:

H = heat loss of the room or building, Btu per hour.

H_1 = heat to be supplied to the reheater coil, Btu per hour.

H_2 = heat supplied tempering coil, or tempering coil and preheater, Btu per hour.

H_3 = heat supplied air washer by water heater, Btu per hour.

H_4 = heat to be supplied booster coil, Btu per hour.

M = weight of air to be introduced into the room or building, pounds per hour.

M_r = weight of recirculated air, pounds per hour.

M_b = weight of air by-passing washer, pounds per hour.

M_o = weight of air drawn in from outside, pounds per hour.

t_o = mean temperature of outside air, degrees Fahrenheit.

t = mean air temperature to be maintained in the room or building, degrees Fahrenheit.

t_1 = mean temperature of the air entering the reheater coil.

t_2 = mean temperature of the air leaving the reheater coil.

t_z = temperature loss in the duct system.

t_y = temperature of the air leaving the duct outlets.

t_x = average temperature of air entering tempering coil.

t_w = temperature of air entering washer.

0.24 = specific heat of air at constant pressure.

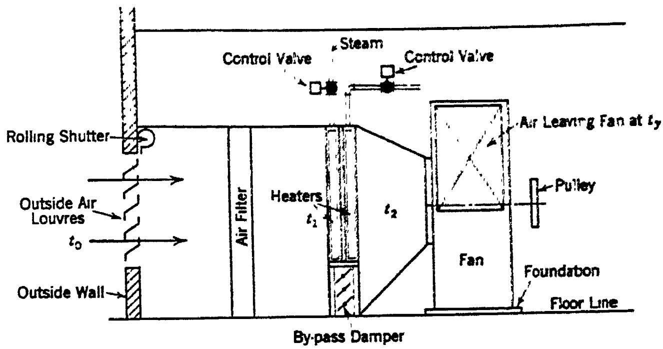

FIG. 3. HEATING UNIT AND FAN ARRANGED FOR OUTSIDE AIR CIRCULATION (*Case A*)

Case A. (Fig. 3) All of the air circulated to be drawn from outside the building, in which case $t_x = t_o$.

$$H_2 = 0.24\ (t_1 - t_o)\ M_o \qquad (7)$$

$$H_1 = 0.24\ (t_2 - t_1)\ M_o \qquad (8)$$

Example 6. The heat loss H for a certain factory building is 700,000 Btu per hour. The mean inside temperature t to be maintained is 65 F. The assumed outside air temperature t_o is 0 F; $t_z = 0$, $t_y = t_2$ and is assumed to be 140 F. The temperature leaving the tempering coil is assumed to be 35 F. Required, H_1 and H_2. From Equation 4,

$$M = \frac{700{,}000}{0.24\ (140 - 65)} = 38{,}889 \text{ lb per hour.}$$

$H_2 = 0.24 \times (35 - 0) \times 38{,}889 = 326{,}667$ Btu per hour.
$H_1 = 0.24 \times (140 - 35) \times 38{,}889 = 980{,}003$ Btu per hour.
$H_2 + H_1 = 326{,}667 + 980{,}003 = 1{,}306{,}670$ Btu per hour.

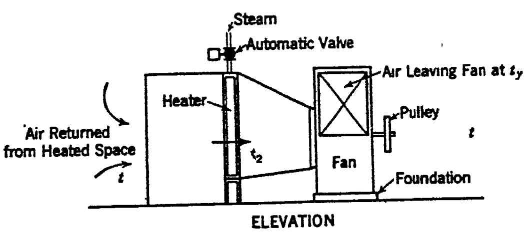

FIG. 4. ARRANGEMENT FOR RECIRCULATION (*Case B*)

Case B. (Fig. 4) All of the air is to be recirculated, in which case $t_1 = t$.

$M_r = 38{,}889$ lb
$H_1 = 0.24\ (t_2 - t_1)\ M_r$
$H_1 = 0.24\ (140 - 65) \times 38{,}889 = 700{,}000$ Btu per hour.

This example illustrates the saving in fuel consumption by the recirculation of the air. The heat to be supplied the apparatus is the same as that required for a direct system of heating and is equal to the heat loss of the building ($H_1 = H$), in the example 700,000 Btu per hour as compared with 1,306,670 for *Case A*.

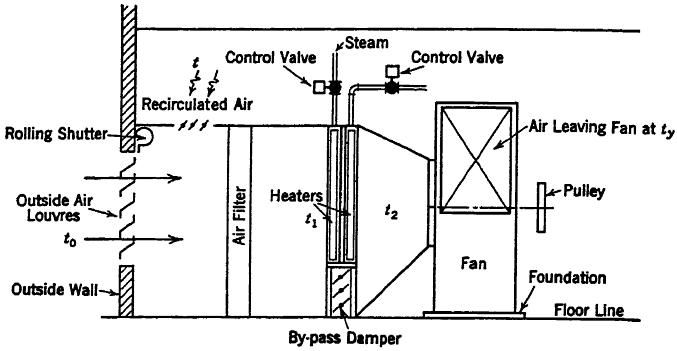

FIG. 5. COMBINATION OF RECIRCULATED AIR AND OUTSIDE AIR (*Case C*)

Case C. (Fig. 5) A portion of the air circulated is recirculated air and the remainder, as may be required for ventilating purposes, is drawn in from the outside. According to Equations 4 and 5,

$$M = M_o + M_r = \frac{H}{0.24\ (t_y - t)}$$

The temperature of the resulting mixture of outside and recirculated air entering the tempering coil is:

$$t_x = \frac{M_o t_o + M_r t}{M} \tag{9}$$

Example 7. Assuming that a positive supply of outside air (d_o = 0.08633) is required for ventilation at the rate of 90,000 cu ft per hour in the preceding example, then M_o = 0.08633 × 90,000 = 7776 lb per hour are required, measured at 65 F.

$$M_r = M - M_o = 38{,}889 - 7776 = 31{,}113 \text{ lb}$$

$$t_x = \frac{7776 \times 0 + 31{,}113 \times 65}{38{,}889} = 52 \text{ F}$$

$$H_1 = 38{,}889 \times 0.24\ (140 - 52) = 821{,}336 \text{ Btu.}$$

This amount of work may be accomplished with one or more banks of heating units, that is, either a single reheater or a tempering coil and reheater.

The three preceding cases refer to installations in which conditioning the air to maintain certain relative humidity requirements does not enter into the problem, as for example, certain types of industrial installations. In practically all modern public buildings, theaters, schools, and in many industrial installations, the ventilating requirements include the provision for washing and humidifying the air delivered to the various rooms of the structure.

In the following cases it is assumed that in addition to maintaining a mean room temperature t, the heating and ventilating apparatus is required to maintain a constant relative humidity in the rooms.

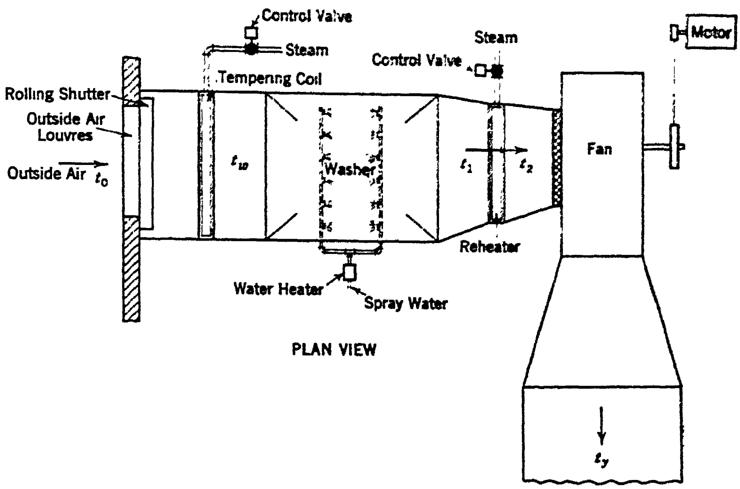

FIG. 6. OUTSIDE AIR CIRCULATED; CONSTANT RELATIVE HUMIDITY IN ROOM (*Case D*)

Case D. (Fig. 6) The maximum relative humidity that may be maintained within the building without the precipitation of moisture on single glazed sash when the outside temperature is 30 F is approximately 35 per cent. If the inside temperature t is 70 F, 35 per cent relative humidity corresponds to a dew-point temperature of 41 F. (See psychrometric chart.)

The installation shown in Fig. 6 contemplates the use of a tempering coil, an air washer provided with a water heater, and a reheater. The tempering coil, one section in depth, warms the incoming air to approximately 35 F to prevent freezing any of the spray water. The air passing through the spray chamber is saturated and leaves at a temperature of t_1 = 41 F.

The heat to be supplied the reheater is:

$$H_1 = 0.24\ (t_2 - 41)M \text{ Btu per hour.}$$

The heat to be supplied the tempering coil is:

$$H_2 = 0.24\ (35 - t_0)M \text{ Btu per hour.}$$

The amount of heat, per pound of air circulated, to be supplied the humidifying washer or humidifier is the difference between the heat content of the assumed dry air entering the washer at a temperature of t_W = 35 F and that of the leaving saturated air at t_1 = 41 F (Table 6, Chapter 1), or:

$$15.657 - 8.397 = 7.26 \text{ Btu per pound of dry air.}$$

The amount of heat required for the washer is:

$$H_3 = 7.26\ M \text{ Btu per hour.}$$

The total amount of heat required by the apparatus is, therefore:

$$H_1 + H_2 + H_3 \text{ Btu per hour.}$$

If a washer having a *humidifying efficiency* of 67 per cent *without water heater* is employed it will be necessary to heat the outside air drawn into the apparatus by means of the tempering and preheater coils to such a temperature that the air in passing through

the water sprays will become partially saturated (adiabatically) having a moisture content per pound of air equal to saturated air at 41 F. If the incoming air is warmed to t_w = 88 F (requiring a two-section-depth heating unit) it will be cooled in the washer to 64 F, with a temperature drop of 88 − 64 = 24 deg.

If the *humidifying efficiency* of the washer were 100 per cent, the air would become adiabatically saturated at 52 F after a temperature drop of 88 − 52 = 36 F. The efficiency of the washer is, however, only 67 per cent, so that the actual temperature drop will be 0.67 × 36 deg or 24 deg, as used.

The heat to be supplied the reheater is in this case $H_1 = 0.24\ (t_2 - 64)\ M$ Btu per hour, and the heat to be supplied to the tempering coil and preheater is $H_2 = 0.24\ (88 - t_0)\ M$. The total heat required by the apparatus is $H_1 + H_2$, no heat being supplied to the washer.

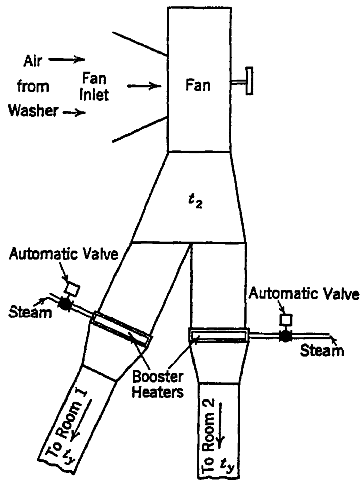

FIG. 7. OUTSIDE AIR CIRCULATED; CONSTANT TEMPERATURE AND RELATIVE HUMIDITY MAINTAINED IN EACH ROOM (*Case E*)

Case E. (Fig. 7) The temperature t_y will ordinarily be different for each room. With H and M fixed, $0.24\ (t_y - t)M = H$, or

$$t_y = \frac{H}{0.24\ M} + t$$

In order to provide the proper temperature for each room, a booster coil is generally installed in each supply duct near the outlet to control the outlet temperature t_y. The amount of steam supplied to these booster units is usually controlled automatically by individual thermostats. The heat required by the booster coils depends on the temperature range through which the air is heated and the quantity of air, or

$$H_4 = 0.24\ (t_y - t_2 + t_z)M \qquad (10)$$

Heat to be Supplied

The amount of heat to be supplied (H') is equal to the sum of the heat requirements of the various heating units and the water heater of the washer, if any, plus the allowance for piping tax. (See preceding *Cases A* to *E*.)

Grate Area, Boiler Selection

The required grate area may be determined by the following formula:

$$G = \frac{H}{F \times E \times C} \tag{11}$$

where

G = required grate area, square feet.
F = calorific value of fuel, Btu per pound.
C = combustion rate, pounds per square foot of grate per hour.
E = boiler and grate efficiency, per cent.

Example 8. Using the data in Example 6, and assuming coal having a calorific value of 12,000 Btu per pound, a combustion rate of 7 lb per square foot, and a performance efficiency of 0.60, and neglecting the piping tax,

$$G = \frac{1,306,670}{12,000 \times 0.60 \times 7} = 26 \text{ sq ft.}$$

Weight of Condensate

The normal weight of condensate to be handled from central fan systems may be estimated by means of the following formula:

$$W = \frac{60\, dQ \times 0.24 \times \Delta t}{h_{fg}} \tag{12}$$

where

W = weight of condensate, pounds per hour.
Q = total volume of air, cubic feet per minute.
Δt = temperature rise of air, degrees Fahrenheit.
h_{fg} = latent heat of steam in the system, Btu per pound.

Ducts and Outlets, Air Filters, Air Washers

The design of the duct system should be based on data contained in Chapter 29. Air washers and humidifiers are described in Chapter 25. For information on air filters, see Chapter 26.

Static Pressure

The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus involved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed:

1. Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 in. of water.
2. A typical oil filter at rated capacity and velocity has a drop of 0.25 in. of water.
3. The loss of one row of a standard make tempering stack equals 0.09 in. water.
4. The loss of one row of a standard make preheater equals 0.10 in. water.
5. A standard humidifier at rated velocity may have a loss of about 0.35 in. water.
6. The loss through one row of a standard make reheater equals 0.12 in. water.
7. A fair assumption for duct losses on a simple system is 0.25 in. water.
8. The static pressure for a nozzle type outlet may be taken as 0.1 in. water.

The sum of these values equals 0.2 + 0.25 + 0.09 + 0.10 + 0.35 + 0.12 + 0.25 + 0.1 = 1.46 in. which is the static pressure against which the system must operate.

Fans and Control

The selection of fans may be based on data contained in Chapter 27 and for motors in Chapter 38. Because centrifugal fans reach their maximum efficiency when working against the resistance offered by the average central fan heating system, they are well adapted to such systems and are generally used. Information on temperature control for central fan systems is given in Chapter 37.

PROBLEMS IN PRACTICE

1 ● Consider a blast heating system handling 10,000 cfm. The resistance to air flow offered by one coil arrangement is 0.9 in. of water and by another coil arrangement is 0.2 in. of water. The fan operates 4000 hours per year and the combined efficiency of motor and fan is 60 per cent. Determine the annual energy saving if the second coil is used.

Difference in system resistance = 0.9 − 0.2 = 0.7 in. of water.

$$\text{Reduction in power input} = \frac{10{,}000 \times 0.7}{6356 \times 0.60} = 1.83 \text{ hp.}$$

Annual energy saving = 1.83 × 0.764 × 4000 = 5480 kwhr.

2 ● What saving results from recirculating some of the room air and reducing the amount of outside air?

Because outside air must be heated to room temperature, reducing the amount of outside air produces a proportionate saving in heat or fuel.

3 ● What items make up the total heating load in a central fan heating system?

1. The net heat loss from the conditioned space.
2. The heat required for evaporation of water for humidification.
3. The heat required to raise the temperature of outside air to room temperature.
4. Heat losses from pipes and ducts.

4 ● A group of three drafting rooms, having a total volume of 27,000 cu ft, a transmission loss of 110,100 Btu per hour, and an infiltration loss of 34,200 Btu per hour on the basis of 0 F outdoors and 70 F room temperature, is to be heated by a recirculating hot blast heating system with air entering the rooms at 116 F. How many cubic feet per minute, measured at 70 F, will be required?

Substitute in Equation 3. H = 110,100 + 34,200 = 144,300 Btu per hour; t_y = 116 F;

$$t = 70\text{ F};\ Q = \frac{144{,}300}{60 \times 0.07492 \times 0.24\,(116 - 70)} = 2900 \text{ cfm}$$

5 ● In the preceding question, if the warm air loses 4 F between heater and rooms, how many pounds of steam per hour at 1-lb gage will the heating sections condense?

Substitute in Equation 12. Q = 2900 cfm, from solution of Question 4; Δt = 116 + 4 − 70 = 50 F; h_{fg} = 968 Btu, from steam table in Chapter 1.

$$W = \frac{60\,dQ \times 0.24 \times \Delta t}{h_{fg}} = \frac{60 \times 0.07492 \times 2900 \times 0.24 \times 50}{968} = 161.8 \text{ lb per hour.}$$

Chapter 22

CENTRAL SYSTEMS FOR COOLING AND DEHUMIDIFYING

Classification of Systems, Spray and Surface Type Dehumidifiers, Designing the System, Zoning, Location of Apparatus, Air Temperature Leaving Room Inlets, Calculations and Selection of Apparatus, Quantity and Temperature of Air Required, Heat Removed by Apparatus, Reheating Dehumidified Air, By-Pass System

CENTRAL systems, equipped for cooling and dehumidifying, are used principally in the air conditioning of theatres, restaurants, office buildings, or other places where people gather, and in manufacturing establishments where air conditions have an important influence on the quality of product or rate of production. A central cooling and dehumidifying plant is one in which the fans, dehumidifiers, and other related apparatus are assembled in suitable apparatus rooms from which supply and return ducts lead to the conditioned spaces. The design of such systems is considered in this chapter, while in Chapter 21 Central Systems for Heating and Humidifying are described. Air conditioning for industrial processes is considered in Chapter 33. A discussion of the dehumidifying equipment only, will be found in Chapters 24 and 25.

CLASSIFICATION OF SYSTEMS

Dehumidification or cooling of air may be accomplished by several methods, and by use of many heat transfer media. Most central station comfort air conditioning systems employ cold water or the direct expansion of a refrigerant in either spray type or surface type equipment to accomplish the required cooling and dehumidification. Hence this chapter will be concerned mainly with the design of such systems.

Two other methods of summer air conditioning are used to some extent. In regions where the summer wet-bulb temperature is low (see Chapter 8, Table 1), evaporative cooling can be used. A spray type unit is employed, with recirculation of the spray water and usually a supply of 100 per cent outside air. The dry-bulb temperature of the air is reduced but the relative humidity of the air is increased, as the air passes through the sprays and its sensible heat is converted into latent heat. The wet-bulb temperature remains constant, and for comfort conditioning it is advisable to have the final dry-bulb temperature a few degrees higher than the wet-bulb.

Another method of summer air conditioning is that in which the air is passed over a dehydrating agent, and then the dry-bulb temperature is lowered to the proper level. This latter step is usually accomplished by means of water, and the water temperature need not be as low as is required for dehumidification by condensation.

For the common method of dehumidifying the air by cooling it below the dew-point, either the air is conducted through a low temperature

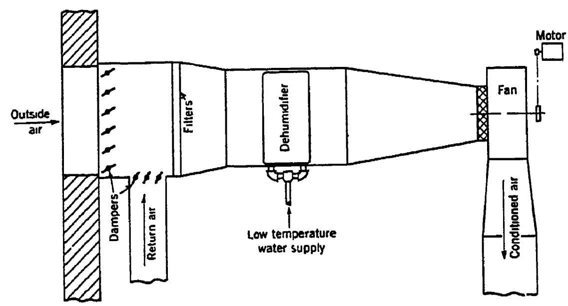

FIG. 1. SIMPLE DEHUMIDIFYING APPARATUS

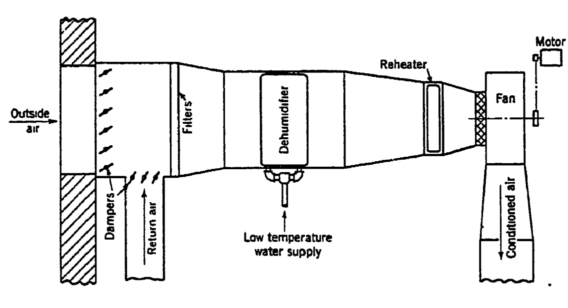

FIG. 2. DEHUMIDIFYING EQUIPMENT WITH REHEATER

liquid spray, or it is directed over surface coils through which cold water or evaporating refrigerant is circulated. The same principle governs the operation of both spray and surface type dehumidifiers, viz.: *The dew-point temperature of the air leaving the dehumidifier should be such that when the air is raised to room conditions it will have absorbed the latent heat load of the rooms.*

The arrangement of a simple summer air conditioning plant is shown in Fig. 1. The plant may be designed to condition 100 per cent outside air, 100 per cent return air, or a mixture of outside and return air. If the

system is intended solely for summer conditioning, the apparatus will consist essentially of a dehumidifier of spray or surface type, filters, fan and motor, duct work for outside air, return air and conditioned air supply, air inlets and outlets and suitable controls. For the spray unit or for water coils a pump will be required, and some form of cooling equipment must be installed unless a supply of sufficiently low temperature natural water is available. In many cases a reheater will be necessary, as described in the next paragraph. Frequently a central air conditioning system is designed for year-round service. This means that properly sized heaters and humidifiers, with their respective controls, must be added. With few exceptions, systems designed to meet summer capacity requirements will have ample capacity for winter and intermediate season conditioning.

In lowering the dew-point temperature to enable the air to carry off the latent heat load, the dry-bulb temperature may be lowered excessively. The air must then be reheated before being delivered to the rooms.

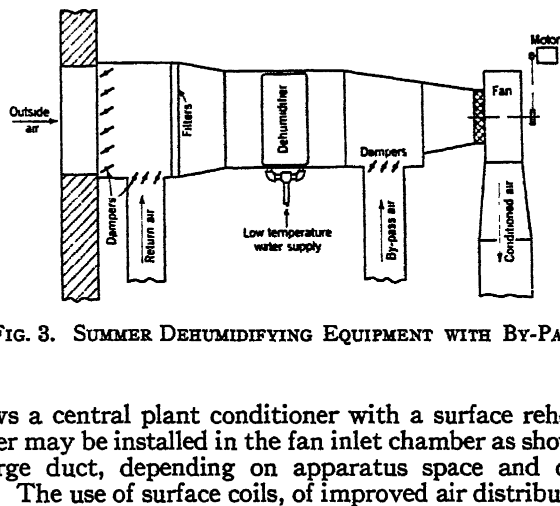

FIG. 3. SUMMER DEHUMIDIFYING EQUIPMENT WITH BY-PASS

Fig. 2 shows a central plant conditioner with a surface reheater added. The reheater may be installed in the fan inlet chamber as shown, or in the fan discharge duct, depending on apparatus space and other design conditions. The use of surface coils, of improved air distribution systems in the rooms, and of smaller refrigeration units with automatic control, are all tending to make the use of reheaters less essential.

Another method for raising the dry-bulb temperature of the conditioned air supplied to the rooms, is to by-pass some of the return air[1] so that it enters on the downstream side of the dehumidifier, as shown in Fig. 3. This method of reheating the air may be more economical in operation than using a surface reheater where such a reheater cannot be supplied with waste heat.

In some cases the main supply fan delivers the dehumidified air to several other fans rather than to the conditioned space directly. These booster fan units may be arranged to deliver a mixture of conditioned air

[1]Patents exist covering the use of the by-pass for cooling and dehumidifying systems.

and return air as shown in Fig. 4, or they may be equipped with reheaters and take in 100 per cent dehumidified air.

The systems illustrated in Figs. 1 to 4 may have either spray dehumidifiers or surface coils, and the latter may use either cold water or direct expansion refrigerant. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver saturated or nearly saturated air, tends to simplify the control problem. In this case the dry-bulb temperature is also the dew-point, and hence a dew-point control can be arranged by using a simple duct thermostat. Other advantages of the spray system are that it may be used for hu-

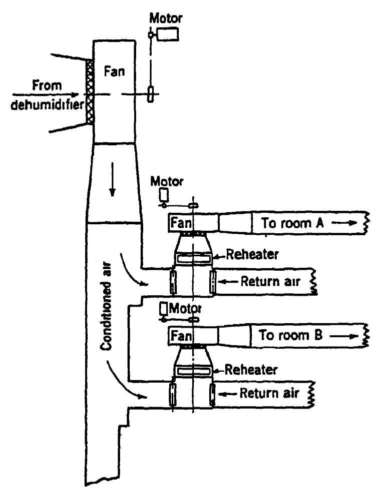

FIG. 4. CENTRAL DEHUMIDIFYING PLANT AND LOCAL RECIRCULATING FANS

midifying in winter or for evaporative cooling when the outside wet-bulb temperature is low.

Surface coil dehumidifiers seldom deliver saturated air. A wet-bulb depression of 2 to 5 F (or more) is usual, and with this higher dry-bulb temperature (for a given dew-point), reheating may be unnecessary as indicated in Table 1. Where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs, but some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream. Therefore, local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The performance of a surface type dehumidifier is affected by air velocity, refrigerant velocity, temperature and moisture content of the entering air, piping arrangement and coil design, and these variations must be taken into account in the design of a system which includes a surface-type unit.

TABLE 1. ROOM HEAT LOAD RATIOS FOR TYPICAL SUMMER COMFORT CONDITIONING

ROOM HEAT LOAD RATIOS[a]	TYPICAL CLASSES OF ROOM SERVICE OR LOAD				
	No. Occupants or Sources of Vapor	Private Office or Residence	Restaurant or Crowded Office	Auditorium at Capacity or Crowded Restaurant	Ballroom at Capacity
$\frac{\text{Sensible Heat}}{\text{Total Heat}}$	1.00	0.90	0.80	0.70	0.60
$\frac{\text{Total Heat}}{\text{Sensible Heat}}$	1.00	1.11	1.25	1.43	1.67
$\frac{\text{Latent Heat}}{\text{Total Heat}}$	0	0.10	0.20	0.30	0.40
$\frac{\text{Total Heat}}{\text{Latent Heat}}$	....	10.00	5.00	3.33	2.50
$\frac{\text{Sensible Heat}}{\text{Latent Heat}}$	....	9.00	4.00	2.33	1.50
$\frac{\text{Latent Heat}}{\text{Sensible Heat}}$	0	0.11	0.25	0.43	0.67
Dry-bulb Temperature of Air at Room Inlets, to Maintain Typical Room Conditions of 80 F Dry-bulb, 50 per cent Relative Humidity					
Air entering saturated[b]	60.0	58.6	56.5	53.0	35.0
Air entering with 4 F wet-bulb depression	66.5	65.4	64.1	61.8	56.0
Air entering with 8 F wet-bulb depression	72.6	72.1	71.6	70.5	68.0

[a]The overall heat load ratio for the dehumidifier will be different from the heat load ratio for the room. The extent of the difference will depend on the quantity and condition of the outside air used, upon the magnitude of the duct losses, and upon whether or not reheat or by-pass are used.

[b]Typical air conditions leaving the central conditioner are: With spray dehumidifier, 0 to 2 F wet-bulb depression. With surface-type dehumidifier, 1 to 6 F wet-bulb depression. With by-pass or reheat, 4 to 10 F wet-bulb depression.

DESIGNING THE SYSTEM

The general procedure for the design of a central system is as follows:

1. Calculate the sensible heat and latent heat gains for each room or space to be conditioned. (See Chapters 6, 7 and 8).
2. Establish the temperature of air leaving the supply inlets.
3. Calculate the quantity of air to be circulated.
4. Estimate the temperature rise in the duct system.
5. Determine the volume of outside air to be introduced. (See Chapter 3).
6. Calculate the heat to be removed by cooling and dehumidifying apparatus, and the type and arrangement of apparatus to be used.
7. Calculate the size of the reheating equipment, if any.
8. Select cooling and dehumidifying equipment, and refrigerating and reheating equipment, from manufacturers' data and performance curves.
9. Design the air filtering and distribution system, the air outlets and inlets. (See Chapters 26, 28 and 29).
10. Calculate the total static pressure of the system.
11. Select the fan, motor and drive. (See Chapters 27 and 38).
12. Select the pump and motor.
13. Design the control system. (See Chapter 37.)

ZONING THE SYSTEM

The foregoing general outline of procedure will prove satisfactory for the smaller and less complex installations. However, when dealing with air conditioning systems for large buildings, after a proper analysis has been made of the conditions to be maintained and the heat loads encountered, it is generally considered good practice to divide the complete job into a number of suitably sized units. In some cases a unit per floor or group of floors may complete the design satisfactorily, whereas in others it may be advantageous to have separate units for each of the various outside exposures of the building. The heat loads on inside rooms are apt to be less variable since the fluctuations of the outside weather conditions are not directly involved. Where the floor area is large in relation to the outside wall exposure, it is obvious that special provision must be made for the variable load to which the outside exposures are subjected. Such conditions often result in the natural zoning or segregation of rooms having similar exposures and internal heat loads. Variations in the hours of occupancy in different portions of a building also frequently require careful zoning for successful operation.

LOCATION OF APPARATUS

Availability of space for apparatus and duct work is of primary importance when selecting the type of system for a given design. In general, for large installations, the refrigeration equipment, because of its size, weight, and operating characteristics, is located in the basement along with the boilers, fire pumps, and other equipment. The air conditioning apparatus is generally located where clean outdoor air is readily available, the designer bearing in mind that supply and return air ducts, steam connections, water and drain connections, and electrical connections must be made to the equipment proper.

AIR TEMPERATURE LEAVING ROOM INLETS

In comfort conditioning applications, air has been distributed from properly designed inlets without producing drafts at temperatures varying from approximately 5 to 30 F below the required room temperature. Factors influencing the design and selection of air inlets are: ceiling height, contour of ceiling, length of blow, and temperature and quantity of air to be distributed. Most summer conditioning installations are designed to supply the air to the conditioned space at from 8 to 18 F below room temperature. Recently the use of specially designed nozzles has indicated the possibility of reducing the air quantity necessary to dissipate a given heat load by introducing the air into the room as much as 30 F below room temperature. Directional flow inlets which spread the air fanwise permit lower inlet temperatures than single direction inlets. Comfort conditioning systems employing differentials greater than 18 F require special consideration and design experience because high pressure inlets or nozzles are usually used. Further, care must be taken to allow a sufficient air quantity under all load conditions to insure good distribution. If winter heating, as well as summer conditioning, is to be accomplished by the same distributing system, the design of the

inlets will be influenced as discussed in Chapter 21. Industrial systems in which drafts are not objectionable usually employ a temperature differential equal to the dew-point depression.

CALCULATIONS AND SELECTION OF APPARATUS

When the cooling loads in the rooms to be served have been calculated as outlined in Chapter 8, they are combined to obtain the total room load. However, all loads must be calculated in two parts: (1) the sensible heat or dry load, and (2) the latent heat or moisture load. For convenience it is customary to state this division of loads by a ratio, as for instance the ratio of sensible heat load to total load. Unfortunately there is as yet no uniform practice in the statement of this ratio, and hence in Table 1 all the common ways of stating the load ratio are given. It should be noted that the heat load ratio for the dehumidifier is not exactly the same as the heat load ratio of the room except in the case of 100 per cent recirculation and zero reheat.

The heat load ratio of a room depends upon its occupancy as well as upon the heat transmitted through its walls and windows. This is approximately indicated in Table 1. Since human occupants are one of the greatest sources of latent heat or moisture load, this load is frequently a minimum when there is a large room space per occupant and the occupants are not doing physical work.

Examples of the solution of a typical problem of treating air to produce room conditions of 80 F dry-bulb and 50 per cent relative humidity are also given in Table 1. In these examples it is assumed that as the air is discharged into the room and diffuses with the room air, it is required to absorb sensible and latent heat in the ratio indicated, and that its final condition after absorbing this heat is the *room condition* of 80 F dry-bulb and 50 per cent relative humidity. Table 1 deals with the room only, and the heat load ratios for the dehumidifier and the temperatures leaving the same will not be identical with those given for the room. However, if the heat gains in the duct work have been included as part of the room load, the dry-bulb temperatures in Table 1 will be those at the discharge of the central conditioning apparatus. To obtain the total heat load on the dehumidifier, and its corresponding heat load ratio, the load due to outside or ventilating air must of course be added. The significance of these statements is illustrated in the examples following.

Quantity and Temperature of Air Required

The quantity of air to be circulated is usually determined on the basis of the sensible heat load, although in some cases the air quantity will be determined by the latent heat load, or by the air distribution or ventilation requirements.

Example 1. A room is to be maintained at a dry-bulb temperature of 80 F and a relative humidity of 55 per cent, (68.5 F wet-bulb, 62.5 F dew-point, 85.5 grains of moisture per pound). The sensible heat gain in this room is 100,000 Btu per hour, and the latent heat gain is 33,000 Btu per hour, or a heat load ratio, sensible to total, of 75 per cent. A temperature differential of 12 F between the room air and the conditioned supply has been selected, i.e., the air is to be supplied at 68 F dry-bulb. Find the quantity and condition of the air supply required.

Solution. From a table of air properties, (Chapter 1, Table 6), it is found that the sensible heat content of air at 80 F is 19.19 Btu per pound, and that of air at 68 F is 16.31 Btu per pound. Hence the sensible heat load which can be absorbed by 1 lb of air is 19.19 − 16.31 = 2.88 Btu. Then the total air quantity required is 100,000/2.88 = 34,700 lb per hour. Expressed in terms of standard air at 0.075 lb per cubic foot, this is: 34,700/(60 x 0.075) = 7,720 cfm. Since a latent heat load of 33,000 Btu per hour must be absorbed by 34,700 lb of air per hour, the latent heat to be absorbed per pound of air is: 33,000/34,700 = 0.952 Btu. The latent heat per pound of *vapor* is approximately 1060 Btu (from steam tables), and since 1 lb = 7,000 grains, the moisture to be added to each pound of air is: (0.0952 x 7000)/1060 = 6.3 grains. Hence the dew-point temperature of the conditioned air supply will correspond to 85.5 − 6.3 = 79.2 grains per pound, and from the psychrometric chart or tables this is found to be 60.7 F dew-point. At a dry-bulb temperature of 68 F this corresponds to a wet-bulb temperature of approximately 63.3 F, or a relative humidity of 78 per cent. Summarizing, the required air supply is: 7,720 cfm at 68 F dry-bulb and 63.3 F wet-bulb.

In the previous example and solution, no consideration was given to the type of air conditioner to be used. *The solution is dependent on room conditions only, and the method is generally applicable regardless of the heat load ratio or the size of the system.* Of course if the air quantity or the temperatures obtained in the solution are not practicable for the actual installation, the original selection of a dry-bulb temperature of the inlet air may be changed as required, providing the design of the distribution system is modified accordingly. A higher dry-bulb temperature of the air supply, i.e., a smaller temperature differential, will call for a larger quantity of air and a larger wet-bulb depression at the supply inlets.

Heat Removed by Apparatus

The total cooling and dehumidifying load will depend on the total room load, the duct losses, the outside air or ventilation load, and the amount of reheat, if any. The necessity for reheat will in turn be determined by the type of system, and by the dry-bulb temperature required at the room supply inlets.

Example 2. To maintain a room at 80 F dry-bulb and 55 per cent relative humidity requires a conditioned air supply of 7,720 cfm at 68 F dry-bulb and 63.3 F wet-bulb temperature (see Example 1.). The air is to be conditioned in a central plant unit of the type shown in Fig. 2. The conditioned air is to consist of 30 per cent outside air (2320 cfm) and 70 per cent recirculated air (5400 cfm). The outside air enters the conditioner at 95 F dry-bulb and 75 F wet-bulb temperature, and the return air is assumed to enter at 80 F dry-bulb and 68.5 F wet-bulb, (neglecting radiation and duct losses). Find the total refrigeration load, and the air conditions entering and leaving the dehumidifier, for both spray and surface type units.

Solution. The simplest method of solution is on the basis of wet-bulb temperatures and total heats, calculating the return air and outside air loads separately. Before this can be done, the air conditions at the exit of the dehumidifier must be determined. It is known that the dew-point of the conditioned air must be 60.7 F. The dry-bulb temperature at the exit of the dehumidifier will then depend on whether sprays or coils are used, and on the design of each. Assume in this case that the spray dehumidifier saturates the air, and that the air discharged from the surface dehumidifier has a 3 F wet-bulb depression. (These are common assumptions, but other values may be obtained, depending on the designs). Air conditions at the discharge of the dehumidifier are then: For the spray dehumidifier, dry-bulb = wet-bulb = dew-point = 60.7 F. For the surface type dehumidifier, dry-bulb = 65.5 F, wet-bulb = 62.5 F, dew-point = 60.7 F. The total refrigeration load for each type of dehumidifier may be calculated as follows:

Spray Type Dehumidifier. 2320 cfm of outside air cooled from 75 F wet-bulb (38.46 Btu per pound, total heat content), to 60.7 F wet-bulb (26.86 Btu per pound, total heat content.) Refrigeration load: 2320 × 0.075 × 60 × (38.46 − 26.86) = 121,000 Btu per hour. 5400 cfm of return air, cooled from 68.5 F wet-bulb (32.71 Btu per pound),

to 60.7 F wet-bulb (26.86 Btu per pound). Refrigeration load: 5400 × 0.075 × 60 × (32.71 − 26.86) = 142,000 Btu per hour. The total refrigeration load for the dehumidifier is therefore 263,000 Btu per hour or 21.9 commercial tons of refrigeration.

Surface Type Dehumidifier. 2320 cfm of outside air cooled from 75 F wet-bulb (38.46 Btu per pound), to 62.5 F wet-bulb (28.12 Btu per pound). Refrigeration load: 2320 × 0.075 × 60 × (38.46 − 28.12) = 108,000 Btu per hour. 5400 cfm of return air, cooled from 68.5 F wet-bulb (32.71 Btu per pound) to 62.5 F wet-bulb (28.12 Btu per pound). Refrigeration load: 5400 × 0.075 × 60 × (32.71 − 28.12) = 111,500 Btu per hour. The total refrigeration load for the dehumidifier is therefore 219,500 Btu per hour or 18.3 commercial tons of refrigeration.

A summary of the results with the two types of dehumidifiers is given in Table 2.

TABLE 2. COMPARISON OF SPRAY AND SURFACE TYPE DEHUMIDIFIERS FOR EXAMPLE 2

AIR CONDITIONS OR LOAD	SPRAY TYPE DEHUMIDIFIER	SURFACE TYPE DEHUMIDIFIER
Exit Air Conditions:		
Dry-bulb temperature, deg F	60.7	65.5
Wet-bulb temperature, deg F	60.7	62.5
Dew-point temperature, deg F	60.7	60.7
Total refrigeration load, tons	21.9	18.3
Reheat necessary to raise dry-bulb temperature of exit air to 68 F, Btu per hour	60,900	20,800

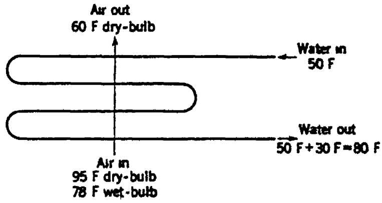

FIG. 5. COUNTER-FLOW SURFACE COOLING DIAGRAM

The design and selection of both spray and surface dehumidifiers is usually made largely on the basis of manufacturers' data, although there are certain general precautions to be observed. Air velocities in spray dehumidifiers are usually limited to 500 or 600 fpm, and the highest temperature of the spray water should be 2 or 3 F below the required exit dew-point. Surface units using cold water should be designed to obtain as near true counterflow as possible as shown in Fig. 5. Face velocities are usually from 400 to 600 fpm, and water velocities should be high enough to obtain good heat transfer, but low enough to avoid excessive pumping costs (preferred range is usually 1 to 3 fps). A close approximation of surface coil area can be made on the basis of the sensible heat transfer, if the latent heat load is not more than 25 or 30 per cent of the total. In this calculation the dry coil heat transfer coefficients are used, and the computation is the same as that used in selecting an air heating coil. If the coil loads, the entering air conditions, and the refrigerant and air velocities are specified by the designer, then the refrigerant temperature and the depth of coil to be used are dictated by the coil design, and cannot be arbitrarily selected. Surface coil performance is greatly affected by the refrigerant temperature, and if the refrigerant temperature

:an be varied, as in a cold water system, the performance of the unit may o a certain extent be varied with the weather and load changes.

REHEATING DEHUMIDIFIED AIR

Table 2 (Example 2) indicates that the amount of reheating necessary o raise the dry-bulb temperature to the selected room inlet condition of ;8 F, is greater in the case of the spray type dehumidifier than with the urface-type unit. The reheating process is a simple addition of sensible ıeat, and the calculation for the spray-type unit is as follows:

Example 3. Find the capacity of the reheater required in connection with the spray lehumidifier in Example 2.

Solution. The heat to be supplied in Btu per hour by reheaters equals sensible heat to aise air from 60.7 to 68 F = 0.24 × 7720 × 0.075 × 60 × (68 − 60.7) = 60,900. urface coils of this capacity must therefore be selected.

3y-Pass System

Example 4. The total sensible heat gain in a restaurant when the dry-bulb temperaure is held at 80 F is 200,000 Btu per hour. The conditioned space shown in Fig. 6,

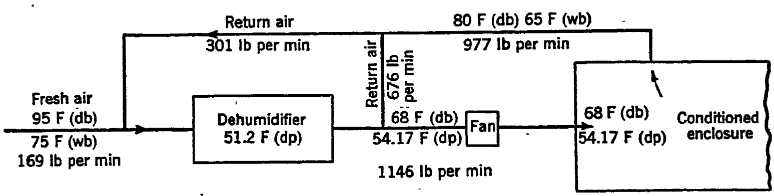

FIG. 6. DIAGRAM OF BY-PASS METHOD

ılso has a moisture gain of 384,000 grains per hour, and an outside air ventilation requirenent of 2250 cfm. Assume (as in Examples 1 and 2), a 12 F dry-bulb temperature lifferential between the entering air and the room temperature, which is the same as ıssuming the dry-bulb temperature of the entering air to be 68 F. It is required to naintain the room conditions at 80 F dry-bulb, 65 F wet-bulb, 56.5 F dew-point. Cal:ulate the air capacity of the system, the amount of return air to be by-passed, and the lew-point temperatures at room inlet and dehumidifier outlet.

Solution. In this system, instead of passing all of the air through the dehumidifier or cooling and dehumidifying, a portion of the return air is mixed with the conditioned ıir at the leaving end of the dehumidifier. The mixture is proportioned so that the esultant conditions are those required at the room inlets, (neglecting losses).

As in Example 1, the sensible heat load which can be absorbed by one pound of air is 2.88 Btu, and the total air quantity required is then: 200,000/2.88 × 60 = 1146 lb per ninute, or about 15,300 cfm.

From Table 6, Chapter 1, the grains per pound of saturated air at 56.5 F is 68.0. The latent heat load is already expressed in terms of grains of moisture, hence the mois:ure to be added to each pound of air is: 384,000/1146 × 60 = 5.6. This gives the noisture content in the entering air which equals 68.0 − 5.6 = 62.4 grains per pound, coresponding to a dew-point at the room inlet, of 54.17 F.

The quantity of air to be dehumidified, the quantity to be by-passed, and the appaatus dew-point temperature may be approximately calculated as follows:

Let

X = percentage of air to be by-passed.

Y = percentage of air to be passed through the dehumidifier.

t_d = apparatus dew-point temperature, degrees Fahrenheit.

The quantity X of 80 F air must mix with the quantity Y of dehumidified air to produce air with a resultant 68 F dry-bulb temperature. Also, X quantity of air at 56.5 F dew-point must be mixed with Y quantity of dehumidified air to give a resultant dew-point temperature of the mixture of 54.17 F. It is assumed that the air passing through the dehumidifier is saturated.

Solving simultaneous equations,

$$80.0X + Yt_d = 68.00 \qquad (1)$$
$$56.5X + Yt_d = 54.17 \qquad (2)$$
$$23.5X + 0 = 13.83$$

$$X = \frac{13.83 \times 100}{23.5} = 59 \text{ per cent, air by-passed.}$$

$$Y = 100 - X = 41 \text{ per cent, air passed through dehumidifier.}$$

The second step is to determine the apparatus dew-point temperature, Substitute X in either Equation 1 or Equation 2, and solve for t_d:

$$80 \times 0.59 + t_d \times 0.41 = 68$$

$$t_d = \frac{68 - 47}{0.41} = 51.2 \text{ F, the apparatus dew point.}$$

PROBLEMS IN PRACTICE

1 ● What is meant by the term evaporative cooling?

Evaporative cooling, or adiabatic saturation of the air, is only effective when the air to be cooled is very dry. It is accomplished by passing the air in an unsaturated condition through a water spray which evaporates a part of the water at the expense of the sensible heat. In this adiabatic transfer the total heat content of the air remains constant while the dew point rises and the dry-bulb falls until the air is saturated.

2 ● In central systems for cooling and dehumidifying what factors fix the quantity of air required?

The weight or volume of air required depends wholly on the sensible heat gain in the room conditioned and on the difference between the dry-bulb temperature of the air at the room inlets and the dry-bulb temperature maintained in the room.

3 ● In central systems for cooling and dehumidifying can the dry-bulb temperature change be fixed arbitrarily?

No, because the change depends on factors at both the conditioner and the room. At the conditioner, temperature of the available water supply may limit the dry-bulb temperature of the leaving air. At the room, the dry-bulb temperature of the entering air may be further limited by: 1. The duct and supply grille arrangement permitted by architectural and structural requirements for the particular space, *e.g.*, ceiling height and obstructions on ceilings, such as beams. 2. The state of activity of the occupants. 3. The velocity at the inlet grille, as limited by noise level requirements. 4. The direction of the jet relative to the occupants.

4 ● What factors determine the dew-point of the air entering the space?

The maximum dew-point desired in the conditioned space, and the moisture gain in the space per unit weight of air supplied.

5 ● Why must the air leaving a dehumidifying type air washer often have its dry-bulb temperature raised before delivery to the occupied zone of room?

The air leaves the dehumidifying air washer saturated at a relatively low temperature which in most cases is lower than the allowable delivery dry-bulb temperature as fixed by factors outlined under Question 3. Also, the air may possibly be carrying a small amount of entrained water which might settle out in the ducts near the washer and cause corrosion difficulties.

6 ● What methods may be used to raise the dry-bulb temperature of the air after it leaves the dehumidifying air washer and before it enters the room?

a. Sensible heat may be added by a reheating method from a source outside the air stream. This method passes all or part of the cold, dehumidified air over steam or hot water coils at the central conditioner or in the ducts, or over electric grids or similar devices. Any available source of sensible heat can be used.

b. A mixing method using sensible heat already in the air stream. In this method the cold, dehumidified air is mixed with air at a higher dry-bulb temperature and the dry-bulb temperature of the resulting mixture is higher than that of the air when it left the conditioner. The air at high dry-bulb temperature is obtained by not passing it through the dehumidifying washer. The mixing may take place at a central conditioner or in the rooms themselves.

c. Combinations of these methods.

7 ● What are the advantages of using counter flow of air and water in surface coolers?

Counter flow results in a higher mean temperature difference than does parallel flow for the same range of air and water temperatures, which means that less cooling surface is required. Counter flow permits higher initial water temperatures and also allows a greater temperature rise for the water. These factors combine to reduce the cost of circulating and refrigerating the cooling water.

Chapter 23

UNIT HEATERS, VENTILATORS, AIR CONDITIONING, COOLING UNITS

Classification of Unitary Equipment and Related Systems, Unit Heaters, Heating Medium, Estimating Heat Losses, Air Temperatures, Output of Heaters, Direction of Discharge, Boiler Capacity, Quietness, Piping Connections, Unit Ventilators, Split and Combined Systems, Vents, Cooling Units, Air Conditioning Units, Heating, Humidifying and Dehumidification, Filtering, Location of Units, Air Distribution, Residential Central System Units, Capacities, Costs, Accessories to Unitary Equipment

IN other chapters, complete descriptions have been given of heating, cooling, ventilating, humidifying, and dehumidifying systems. These descriptions have covered the detailed principles of each and have, in general, described the assembled equipment included in the complete systems. The success of such completely engineered, heating, cooling, and air conditioning systems has inevitably led to the production of smaller factory-assembled equipment employing a majority of the principles of these complete systems. As a result, present day practice involves the use of this unitary equipment in the majority of smaller installations where capacity demands are within the limits of such units. Thus, unit heaters, unit ventilators, cooling units and air conditioning units have come to occupy a place of their own in the industry.

With the growth of this unitary industry, it becomes increasingly evident that there is no sharp line of demarcation, on the basis of capacity, between a *unit* and a *central station* system. Dèfinitions contained in a code, Standard Method of Rating and Testing Air Conditioning Equipment[1], have helped to clarify and identify the various types available.

A *unit* is a factory-made encased assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, humidifying unit, etc. Such units are shipped substantially complete or built and shipped in sections so that the only field work necessary is the assembling together of the sections, without resorting to any field fabrication. A unit of this type may be complete in itself, employing its own direct means of air distribution and sources of refrigeration or heating, in which case, it thus represents a complete self-contained unit. Or it may be coupled with separate means of air distribution such as duct work and outlets, in which case, it will still be considered as a unit system, in dis-

[1]Proposed code prepared by Joint Committee of *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electric Manufacturers Association* and *Air Conditioning Manufacturers' Association*.

tinction to the generally accepted term of a field fabricated central station system. The manufacturer of the unit is responsible for the output and performance of the unit under rated conditions, whereas the contractor installing the complete unitary system is normally held responsible for the complete performance of the system.

Unit equipment justifies its existence due to the following features:

1. Lower cost per unit capacity. Standardized design and volume production makes possible low cost factory assembly thereby eliminating individual design and handling of every part for each installation.

2. Flexibility and mobility of equipment. Unitary equipment can be readily located in existing buildings without the necessity of running large ducts through floors and many partitions. Such equipment can be shifted to meet changing requirements. Tenants may obtain the advantages of conditioning when the entire building is not equipped with a conditioning system. In industrial process work, the flexibility of unitary equipment is also advantageous.

3. Lower installation costs. The fact that the equipment arrives on the job in an assembled condition, coupled with the lesser problems of duct work and connecting piping, materially reduces installation costs.

4. Small capacities. The small capacities available in unitary equipment have brought the advantages of controlled air conditions to a number of small offices, stores, shops, and individual rooms where specially designed and built central system equipment would have been uneconomic.

SUB-DIVISION OF UNITARY EQUIPMENT

For descriptive purposes unitary equipment is sub-divided on a purely functional basis. The following definitions are included in the previously referred to code[2].

1. A *Heating Unit* is a specific air treating combination consisting of means for air circulation and heating within prescribed temperature limits.

2. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.

3. A *Humidifying Unit* adds water vapor to and circulates air in a space to be humidified.

4. A *Dehumidifying Unit* removes water from and circulates air in a space to be dehumidified.

5. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.

6. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining humidity within prescribed limits.

7. A *Heating Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for heating and maintaining humidity within prescribed limits.

8. A *Self-Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements.

9. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

10. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

[2]Loc. Cit. Note 1.

There has grown up in the industry definite branches, in which the engineering and application of the equipment vary quite widely. Thus common acceptance recognizes the following groups of the unitary equipment defined above.

1. *Unit Heaters* consisting of an encased heating surface through which air is forced by means of a fan or blower, located either in or closely adjacent to the heated space, and normally employed only for industrial and commercial applications.

2. *Unit Ventilators* which are similar in principle to unit heaters but are designed to use outside air with or without provision for recirculation of the air. While unit heaters are largely used for commercial and industrial applications, unit ventilators are intended primarily for school, offices and semi-commercial applications.

3. *Cooling Units* which are similar to unit heaters except that a cooling medium is used in place of a heating medium and provision is made to collect and remove the condensate. Cooling units are normally applied to the cooling of products for their preservation or processing (commercial air conditioning) and air conditioning units are used for cooling for comfort.

4. *Air Conditioning Units* which consist of equipment to provide control of heating with humidifying or cooling with dehumidifying, coupled with air circulation: all compactly housed in a single casing.

5. *Miscellaneous Unit Equipment and Accessories* such as filtering equipment, attic fans, humidifying units and special controls.

UNIT HEATERS

A *unit heater* consists of the combination of a heating element and fan or blower having a common enclosure and placed within or adjacent to the space to be heated. Generally no ducts are attached to inlets or outlets, although it is common practice with many unit heaters to equip them with directional outlets or adjustable louvers.

While unit heaters are designed primarily to handle all recirculated air, they may be installed to handle either partial or total outdoor air. Compared with the older method of heating by means of radiation, properly designed and applied unit heaters should:

1. Circulate air in the building at a rapid rate but without objectionable draft.

2. Reduce the temperature differential between the floor and ceiling.

3. Direct the heated air so that uniform temperature distribution be obtained throughout the heated space.

4. Prevent or remove the cold stratum of air commonly found at the floor level.

5. Reduce the number of heating elements required and thereby decrease the cost and extent of the piping necessary.

6. Maintain a closer control of room temperature either manually or by means of simple thermostats.

7. Produce an economy in heating costs resulting from the sum total of the above advantages.

TYPES OF UNITS

There are two major types of unit heaters, propeller fan type and centrifugal housed fan type. The housed fan, high velocity (1500 to 2500 fpm) discharge units with outlets adjustable to deliver air in several directions, are able to project their heating effect over distance of from 30 ft to as much as 200 ft from the unit. This makes possible the location of these units at considerable distances from each other, thus reducing

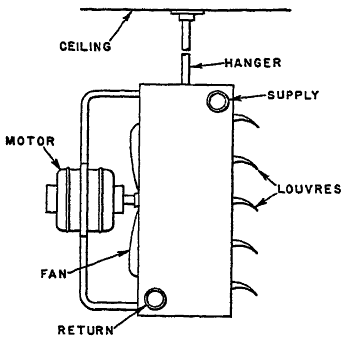

FIG. 1. SUSPENDED UNIT HEATER, PROPELLER TYPE FAN

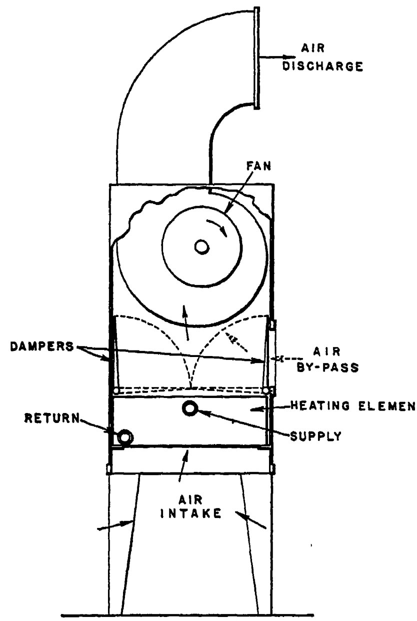

FIG. 2. FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

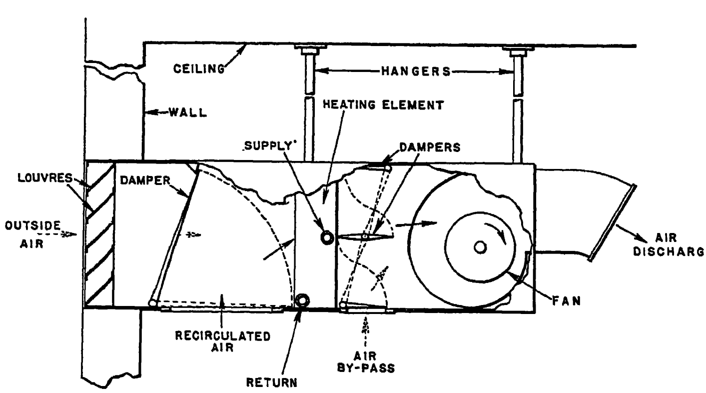

FIG. 3. SUSPENDED TYPE UNIT HEATER, HOUSED TYPE FAN

greatly the piping and loss of floor space due to the heating equipment. Propeller-type units, illustrated in Fig. 1, with outlet velocities of from 300 to 1000 fpm are usually placed from 30 up to 100 ft apart.

Two methods of application of unit heaters are commonly used. Floor mounted units, shown in Fig. 2 are available either with or without the air by-pass, and withdraw the cold air from the floor and discharge the heated air above the working zone. Suspended type units are located in an elevated position withdrawing air from this higher level and discharging the heated air down into the working zone. In closely occupied spaces where direct air drafts into the working zone are not permitted, the floor mounted unit will give more uniform temperature distribution. On the other hand, if opportunity is provided to deliver the heated air from suspended units down into the working zone, excellent temperature distribution is possible. A suspended high-velocity type unit heater connected to an outside air intake with damper to control the volume of ventilation is shown in Fig. 3.

A wide variety of structural designs is available. All employ some form of convector, supplied with either steam or hot water, although occasionally equipped for gas or electric heat. Air is always forced over these convectors by a fan of either the propeller or centrifugal type. Heating surfaces may be in the form of steel pipe coils, non-ferrous tubes or pipes with extended surfaces, cast iron, and pressed or built-up sections of the cartridge or automotive type.

AIR TEMPERATURES [3]

For recirculating heaters with intakes at the floor level, the temperature to be maintained in the room should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. Where suspended heaters are used without any intake boxes extending down to the floor level, a higher entering air temperature should be used than that at which the room is to be maintained.

With suspended unit heaters taking air at some distance above the floor, the temperature variation from floor to ceiling may reach as much as 1 deg for each foot of elevation during the periods when the maximum capacity of the heaters is required. Thus this allowance should be made in calculating the capacity of suspended heaters. High velocity discharge units (blower type) will maintain slightly lower temperature differences than will low velocity units (propeller type). Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. This temperature difference per foot of elevation is less than the corresponding variations for spaces heated by direct radiation.

[3]Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson, and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243).

Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson, and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

TABLE 1. CONSTANTS FOR DETERMINING THE CAPACITY OF UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	**1.000**	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.024	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

OUTPUT OF HEATERS

It is standard practice to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[4]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heat capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Table 1. This table is accurate within 5 per cent.

Unit heaters are customarily rated as free delivery type units. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heat capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design, and speed of the fans employed, so that no specific percentage of reduction can be assigned for all heaters for a given added resistance. In general, however, disc or propeller fan units will have a larger reduction in capacity than housed fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

DIRECTION OF DISCHARGE

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter-directions.

Various types and makes of unit heaters are illustrated in the *Catalog Section* of this edition. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

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[4]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165).

BOILER CAPACITY

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If ventilators are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

QUIETNESS

Fan speed alone is not a measure of relative quietness of fans having different designs and proportions. Quietness is a function of type, diameter, blade form and other variables besides speed, and all these must be considered. In general for a given design, the higher the fan speed, the greater the noise, and centrifugal fans are more quiet than disc or propeller fans.

PIPING CONNECTIONS

Piping connections for unit heaters are similar to those for other types of fan-blast heaters. Typical connections are shown in Figs. 4 and 5. One-pipe gravity and vapor systems are not recommended for unit heater work. For two-pipe closed gravity return systems the return from each unit should be fitted with a heavy-duty or blast trap, and an automatic air valve should be connected into the return header of each unit. Pressure-drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns the same as for vacuum systems and the entire return system be free-vented to the

mosphere, provided all units, drip points, and radiation are properly ıpped to prevent steam entering the returns.

On vacuum or open vented systems the return from each unit should be ted with a large capacity trap to discharge the water of condensation ıd with a thermostatic air valve for eliminating the air, or with a heavy-ıty trap for handling both the condensation and the air, provided the r finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with ɑcuum systems, except that they must be constructed for the pressure ɜed. If the air is to be eliminated at the return header of the unit, a igh pressure air valve can be used: otherwise the air may be passed with ıe condensate through the high-pressure return trap, with some danger f return pipe corrosion and the problem of its elimination at some other oint in the system.

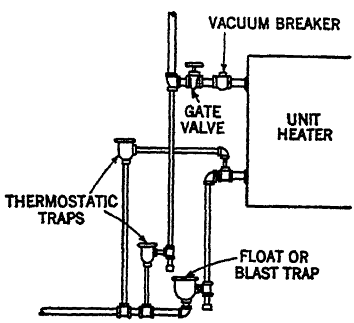

FIG. 4. UNIT HEATER CONNECTIONS WHERE CONDENSATION IS RETURNED TO VACUUM PUMP OR TO AN OPEN VENTED RECEIVER

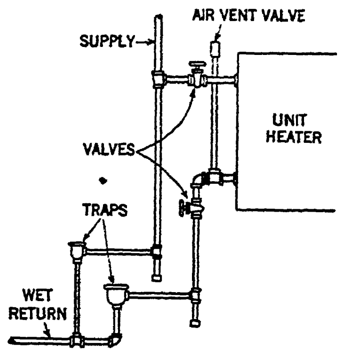

FIG. 5. UNIT HEATER CONNECTIONS WHERE CONDENSATION IS RETURNED TO BOILER THROUGH WET RETURN

OTHER TYPES OF UNITS

All Electric

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. On rare occasions electrical resistances are used as the heating element. These are applied only where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. (See Chapter 40.)

Direct Fired

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm-air type and are equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air in to the space to be heated. As is the case with the steam-fed unit heaters, the gas-fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances

leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified. For permanent installations, it is usually advisable to provide an exhaust duct from the gas-fired unit heaters to remove products of combustion from the occupied space. While this is not necessary in large open industrial plants, in smaller closed rooms, it becomes essential.

Turbine Driven

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

INDUSTRIAL USES

In addition to their prime function of heating buildings, unit heaters may be adapted to a number of industrial processes, such as drying and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye-houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 7.)

UNIT VENTILATORS [5]

Unit ventilators while designed primarily for ventilation must incorporate controlled heating. A typical unit ventilator is illustrated in Fig. 6. They usually consist of a semi-decorative cabinet containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.
3. Adhesive or dry type filters for cleaning the air (optional).
4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together. (No mixing chamber is normally provided where sectional type convectors are used.)
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Discharge grille or diffuser.
9. Temperature control arrangement.

[5] A roof ventilator is sometimes termed a *unit ventilator*. For information on roof ventilators, see Chapter 36.

The primary functions of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air. (See A.S.H.V.E. Ventilation Standards, Chapter 45).

2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.

3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating. (See Chapter 37).

4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.

5. To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary. (Ordinances should be consulted).

6. To perform all its functions without objectionable noise.

7. To clean the air properly.

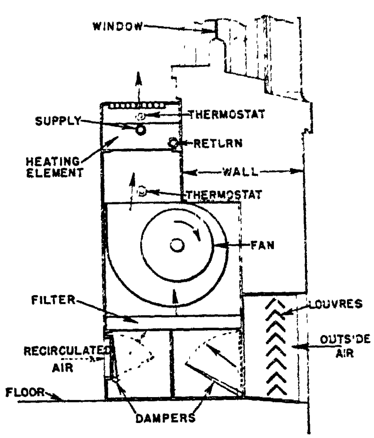

FIG. 6. TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

SPLIT AND COMBINED SYSTEMS

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current,

since the direct radiators will furnish heat, but it permits a carele operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity bei sufficient both for ventilation and for supplying the heat loss. Dire heating surface is omitted altogether. It becomes necessary then that tl fan be running whenever the room is to be heated but this also giv assurance of ventilation, especially if automatic dampers are used in tl air intake from out-of-doors and in the recirculating intake arranged so to give a certain quantity of air from the outside (commensurate wi weather conditions) whenever the unit is operating and after the room heated. The cost of installation of a combined system is usually less tha that of a split system and there is less danger of overheating, but if tl electric energy fails there will be practically no heating.

LOCATION OF UNIT

The location of the unit ventilator in a room is important. Whereve possible it should be placed against an outside wall. It is difficult t obtain proper air distribution if the unit is erected either on an inside wa or in a corner of the room. Standard units discharge the air stream up ward, but for special cases units may be installed to discharge air hori zontally. Units may be set away from the wall or partially recessed int the wall to save space without materially affecting the results. The ai inlet may enter the cabinet at the back at any point from top to bottom

VENTS [6]

The size and location of the vent outlet is important. In many case the sizes for public buildings are regulated by law, but the location of th vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the ven openings nearly equal to that at which the air is introduced into the room thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each

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[6] Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson, and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463).

Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet, and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 279).

classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire, contamination, and other hazards.

CAPACITIES

Unit ventilators are available in air capacities ranging from 450 cfm to 5000 cfm and with corresponding heat capacities (above that required for ventilation purposes based upon an outside temperature of zero and an inside temperature of 70 F) ranging from 15 Mbh to 144 Mbh (1 Mbh = 1000 Btu per hour). Some manufacturers furnish a unit with several heating capacities for each air capacity, thus enabling the engineer to select the unit best adapted to the heating and ventilating load. Typical capacities are given in Table 2[7].

TABLE 2. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

Cubic Feet of Air per Minute	Total Capacity in Square Feet of Equivalent Direct Heating Surface (Radiation)		Capacity Available for Heating the Room in Square Feet of Equivalent Direct Heating Surface (Radiation)		Final Air Temperature (Deg Fahr)
	EDR	Mbh	EDR	Mbh	
600	285	68	95	23	105
750	350	84	115	28	105
1000	455	110	150	36	105
1200	565	136	190	46	105
1500	705	169	235	56	105

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulæ:

When all of the air handled by the unit is taken from the outside,

$$H_t = 0.24\, W\,(t_y - t_o) \tag{1}$$

$$W = d\, 60\, Q \tag{2}$$

$$t_y = \frac{H}{0.24W} + t \tag{3}$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room.
t_o = outside temperature.
t_y = temperature of the air leaving the unit.

[7]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24\, d\, 60\, Q\, (t - t_o) \tag{4}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. H = 24,000; d = 0.075 Q = 1000 cfm; t = 70 F; t_o = 0 F.

Substituting in Equation 4:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70-0) = 99{,}600 \text{ Btu}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o\, (t_y - t_o) + 0.24\, W_i\, (t_y - t) \tag{5}$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.

$$W_o = d_o\, 60\, Q_o \tag{6}$$

$$W_i = d_i\, 60\, Q_i \tag{7}$$

where

d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cubic feet per minute.
Q_i = volume of air taken in from the room, cubic feet per minute.

$$t_y = \frac{H}{0.24\, (W_o + W_i)} + t \tag{8}$$

$$H_t = H + 0.24\, d_o\, 60\, Q_o\, (t - t_o) \tag{9}$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduce from 99,600 Btu to 24,000 Btu, or to about one fourth. Such a unit handling one third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu. Units designed and operated

on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\ W\ (t_y - t) \tag{10}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \tag{11}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

COOLING UNITS

Cooling units as applied to industrial product conditioning and processing are similar in construction to unit heaters except that the heat transfer surface is supplied with refrigeration instead of with steam or hot water. They are normally installed within the space to be served, or at least closely adjacent thereto. Occasionally they are provided to receive outside air in which case this air is invariably filtered or washed to prevent any possible contamination of the product.

Cooling units are provided in two major types similar to unit heaters, either floor mounted with housed fan, or suspended with propeller type fans. Normally, air outlet velocities are lower than for heating, due largely to the effect of high velocities on the product. Cooling units are normally of the free delivery type although they occasionally are supplemented with duct work to provide more careful air distribution.

Product cooling originally was accomplished by means of stationary pipe coils. This was later supplemented with the forced fan bunker systems in which air was passed over banks of coils. The present trend in this field is toward a more accurate control of both temperature and humidity, thus placing these units in the classification of complete air conditioning units as discussed in the next section. However, in the majority of these cases dry-bulb temperature is controlled separately from the control of humidity, thus classifying these units as cooling units.

The principal field for cooling units is in cold storage plants, fur storage, fruit packing houses, provision stores, brewery fermentation and stock rooms, candy plants, and other industrial process work. In replacing bunker and wall coils in meat storage plants, cooling units give distinct

advantages in compactness, lower first cost and maintenance expense, ease of defrosting, freedom from drip and the maintenance of sanitary conditions, as well as uniform temperature and humidity under variable load conditions. Cooling units by means of their positive air circulation prevent dead-air spots, frequently objectionable in this industry.

Typical cooling units are shown in Figs. 7 and 8. The former indicates a suspended type cooling unit which may be designed with or without a moisture eliminator. If high air velocities are maintained, an eliminator will be necessary to prevent the drops of moisture from being carried through with the air. The condensation that occurs is collected in a drip pan and removed from the system through a drain pipe. Fig. 8 indicates a typical floor-mounted unit of the housed fan type. The illustration shows a common form of distributing outlet designed to give low outlet velocities together with a controlled distribution. In process work, it is often important that direct air distribution does not impinge on the

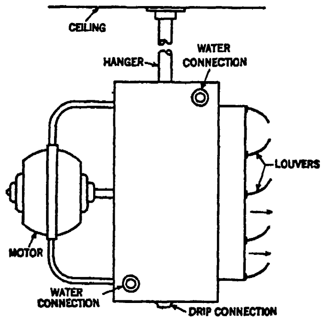

FIG. 7. CEILING TYPE COOLING UNIT

product. Cooling units are normally constructed of galvanized steel or non-ferrous material in order to reduce the corrosive effect of their constant wetted condition.

Cooling units are often called upon to operate in rooms where a temperature below freezing is maintained and low refrigerant temperatures are required. This results in the collection of frost on the heat transfer surface which in turn leads to a rapid loss in capacity and requires eventual defrosting. Such defrosting is accomplished by the following methods:

1. When the room is above freezing the source of refrigeration is cut off and the fan allowed to operate until the unit has defrosted.

2. A reversal of the refrigeration system may be provided and the so-called hot gas defrosting method used. This is accomplished by reversing the flow of the hot gas so that it is delivered directly from the compressor to the evaporator of the cooling unit. As soon as the ice and frost has been melted, the system is again returned to its normal cycle.

3. Where brine is used as a refrigerant, heated brine may be sent through the cooler to remove the ice.

4. When the room is at very low temperatures, warm air defrosting is sometimes used by providing for the admission and removal of warm air from outside the cooled space.

5. The surface may be sprayed with a strong brine solution.

In order to prevent the collection of frost in low temperature rooms where high latent heat loads are present, unit coolers equipped with a constant brine spray are frequently used. These are normally of the housed fan type similar to Fig. 9, but equipped with a pump for recirculating brine over the coil. It is, of course, necessary to strengthen the brine at intervals to maintain a non-freezing mixture.

Ratings of cooling units may be expressed in Btu per hour, or in tons of refrigeration and should specify the quantity, temperature and humidity of the air entering the unit with a stipulated refrigerant temperature

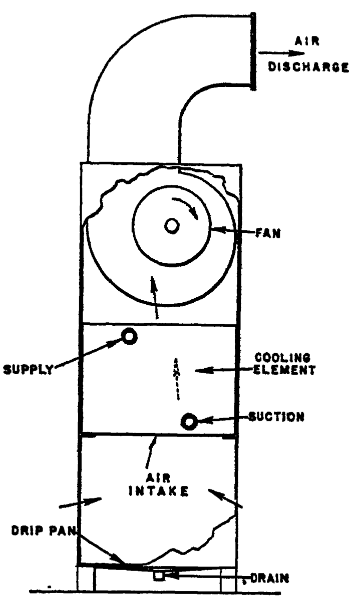

FIG. 8. SURFACE TYPE COOLING UNIT

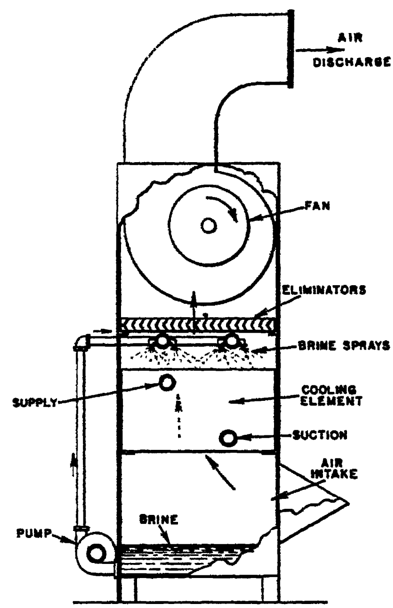

FIG. 9. BRINE SPRAY TYPE COOLING UNIT

within the coil. When chilled water or brine is used, the rate of circulation of the cooling media as well as its entering temperature must be given.

AIR CONDITIONING UNITS

Air conditioning with unit equipment has gained in popularity during the last few years and this type of apparatus now represents the bulk of the production of the industry. It is to be noted that some equipment does not fulfill all of the basic requirements of a true air conditioning unit. True air conditioning equipment involves not only the ability to alter temperature and humidity conditions within the conditioned space, but it must also be able to *control* these conditions.

The means for accomplishing these functions are outlined herewith:

Heating

The normal air conditioning unit derives its heating function from a heating coil, usually of the non-ferrous finned tube type supplied with either steam or hot water. Steam may be supplied directly from a self-contained and built-in oil or gas fired unit, from a separate domestic steam boiler, or even from an outside source of a central heating plant. Hot water is supplied either from a separate hot water boiler or in rare instances from a domestic water heater.

In domestic or household conditioning units, adapted from a warm-air heater to which humidification is added, or possibly all-year-round conditioning, a direct-fired air interchanger is frequently used. The source of heat in this case may be from the combustion of coal, oil, or gas. A wide variety of designs and structures are used. In such direct-fired systems, the bulk and volume of the heat transfer surface is necessarily large in proportion to the rest of the equipment.

Where electric power is low in cost, electric heat has been furnished for air conditioning units either in the form of encased heaters, or open wire heaters. (See Chapter 40.) Radiant electric heaters are seldom used except as their radiant heat is absorbed by some receiving wall and there transmitted to the air in the form of convected heat.

Another method of applying electric heat is by means of the reversed refrigeration cycle, whereby electric energy is used to compress a refrigerant and to deliver the heat of compression, withdrawn from a lower temperature source, to the conditioned space by locating the condensing coils in the air circulation circuit of the air conditioning unit. While this method of heating has gained wide interest, it is practical only in a limited number of applications.

Humidifying

A variety of methods have been used to furnish humidification in winter to air conditioning units. The oldest and best known is by means of a direct spray which is used in many different ways. The simplest system is where the spray water is furnished from a constant water source, such as city water, and is permitted to run to waste. Under such conditions, the spray may be either of the direct atomizing type where, by means of the nozzles, the water is broken into fine particles, or of the so-called target spray type, where a fine stream of water under pressure is caused to impinge upon a flat surface or target. Such methods are normally rather inefficient in the use of water.

In some units, in order to increase the humidifying capacity and to utilize a greater portion of the spray water, the atomized spray is permitted to impinge against a heated surface thereby forcing its evaporation. While this is practical in some instances, there is danger of scale formation where hard water is employed.

One of the simplest methods of humidification in winter is by means of a direct steam spray. This is seldom used in air conditioning units for comfort applications due to the resulting odors. In industrial applications, however, it finds frequent use. The steam is usually introduced to the air through a perforated tube or through some type of porous material.

If a small atomizing spray is not used in comfort conditioning units, the evaporative pan type of humidifier is usually employed. This consists of a container offering as much water surface as possible and equipped with means of heating the water. This heat may be applied either electrically, by steam, hot water, or by circulation of the water from a heated space through the evaporating pan. Since the humidification is accomplished by surface evaporation only, it is essential that the air stream be directed across the surface and that the evaporating surface be large. While this system eliminates the dusting hazard when hard water is used with a spray system, hard water tends to scale the heating surface and results in loss of capacity and the need for frequent cleaning.

The rate of evaporation per unit surface exposed is low, thus it frequently becomes difficult to provide sufficient surface for adequate capacity. The higher the temperature of the water, the lower the relative humidity of the air; the greater the velocity over the surface, the greater is the rate of humidification. The evaporative pan type of humidification limits the water wastage and is usually supplied with water through a float valve. Due to the collection of salts in this evaporating pan such humidifying systems require occasional drainage and cleaning.

Other methods of humidification attempted in air conditioning units are through the use of wetted fabrics, porous eathernware plates, or other capillary surfaces. These methods rely upon the capillary absorption of the moisture up from the liquid level into the portion exposed to the air. They have a tendency to lose their effectiveness due to the resulting deposit of mineral salts at the evaporating surfaces thereby clogging the pores and reducing the contact of the air with the water. Also they frequently become foul and often support bacterial growth.

Cooling and Dehumidification

Those units that employ recirculated water sprays will undoubtedly use such sprays as their means of cooling and dehumidification by furnishing refrigeration to the water in circulation. Occasionally where an adequate source of cold well water is available, this may be used as a direct spray and run to waste.

Other methods of dehumidification accomplished by direct contact with the transfer medium are by means of the so-called *adsorption* and *absorption* systems. (See Chapter 24.) It must be recognized that these methods of dehumidification do not in themselves provide cooling. The substance removes the water vapor from the air thereby heating it. This highly dehumidified air may then be cooled either by partial rehumidification or by direct contact with a cooling medium of cold water or direct expansion refrigerant. There are now on the market solid adsorbents such as silica gel and activated alumina. Water solutions of the chlorides of various inorganic elements such as calcium and lithium chloride are the absorbents most frequently used.

Finally a common direct means of cooling and dehumidification is through the use of ice. In such units the ice is brought into as intimate contact as possible with the air handled. Provision is made for the removal of the moisture as rapidly as it is formed from the melting of the

ice. Ice is also used to cool water which is circulated through the sprays.

In conditioning units, the use of surface cooling is probably more common than direct spray or other direct transfer means. The type of surface employed may, of course, be cast or fabricated from tubes. In present day practice finned tubes or plate fins through which tubes are passed form the most generally used cooling surface. The detailed fabrication of this surface and the arrangement of the tubes will depend largely upon the type of refrigerant for which it is intended.

The simplest construction is where chilled water or brine is used as the refrigerating medium. With direct expansion refrigerant it is usually necessary to provide a special arrangement of headers so that proper distribution of refrigerant through all the surface is obtained. In some cases, ordinary brine coils can be used when operated as a flooded refrigerant system. In some units a combination of a direct spray and a refrigerant surface is used, the spray being directed against the surface. Such systems claim the advantage of air washing together with the maintenance of a clean and effective cooling coil.

It should be noted that when surface coolers are used, adequate protection in the form of filters or at least lint screens are necessary to prevent fouling of the surface from the air borne dirt. Surface not so protected frequently becomes completely matted with lint, grease, and similar dirt.

The sources of refrigeration used with these surface type conditioning units are discussed in Chapter 24. However, they may be divided into the following groups:

1. Direct expansion refrigerant in which the liquid refrigerant is evaporated within the coils of the unit. The vapor from these coils may be recompressed in centrifugal, rotary, or reciprocating type compressors, and the refrigerant again returned to the evaporator coil.

2. Indirect refrigeration by means of:
   a. Cold well water.
   b. Cold city water.
   c. Artificial refrigerated water provided by direct expansion of refrigerant in a water cooler, direct steam jet refrigeration, or by the melting of ice.

Filtering—Air Cleaning

A variety of methods are employed as a means of controlling air purity. In unit systems where filtering alone is considered satisfactory, the degree of filtering varies widely and in proportion to the actual needs. If the air is chiefly recirculated with but little outside air used for ventilation, filtering requirements are largely limited to keeping the coils in a clean and operable condition. Thus such units are frequently furnished with simple lint screens of low resistance and formed of moderately close meshed wire. Where outside air is used for ventilation, more complete filtering of dust particles is necessary and for this purpose, there are a large number of filters available on the market. Some of these filters are of the so-called *throw-away* type, constructed of inexpensive material so that when they become dirty or clogged they may be thrown away and replaced with new ones. All of these filtering methods are described in detail in Chapter 26.

Ventilation

Inasmuch as air purity is one of the factors that constitute true air conditioning, ventilation or the introduction of outside air is an essential part of any air conditioning unit or system. While a unit that recirculates all its air capacity is still considered an air conditioning unit, the better type system provides for the introduction of a certain proportion of outdoor air. In some instances one of several units may operate entirely on outside air, while in other cases only a portion of the air handled by the unit is drawn from out-of-doors. In such cases a damper is provided either in the unit or in the duct connections for controlling the proportion of outdoor air.

Location of Air Conditioning Units

The characteristics of the conditioned space, the building construction, the type of system employed, the duct connections as well as the source of power, piping and refrigeration influence directly the location of air conditioning units.

Primarily a unit is either of the portable or fixed type. The portable unit is usually a simple self-contained air conditioning or cooling unit either with or without ventilation, but includes the condensing unit. If the condensing unit is of the air-cooled type, the unit must be located adjacent to a window or other source of outside air. On the other hand, if it is of the water-cooled type, its location should be convenient to sources of water and drainage. Portable units are invariably located within the conditioned space.

Non-portable units, or units of fixed location may or may not be located within the occupied space. Naturally units that are located in such spaces must be built with decorative cases in order to harmonize with the surroundings. Such units are normally of comparatively small capacity varying from a fraction of a ton up to as much as five or six tons. Many conditioning units and particularly those of the larger sizes are located externally to the occupied and conditioned space and are connected thereto by means of delivery and return ducts. Such an arrangement permits the location of the conditioning unit convenient to either the sources of refrigeration or outside air. It frequently permits the use of the basement or of space less valuable than that on the level or floor of the occupied zone. Oftentimes the same type of unit may find application in an exposed position for one job and in a concealed location for another. Thus it can be seen that it is not possible to define a unit merely on the basis of its location. Frequently conditioning units are built into the structure or into the architectural design of a room so that they are entirely concealed except for the discharge and return grilles or openings which are designed so as to correspond to the decorative scheme of the room.

Air Distribution

With portable units or units exposed within the conditioned space, the distribution is usually through grilles or louvres built entirely into the equipment. The discharge of the air from this unit, in general, should be

upward immediately at the unit, with sufficient horizontal component to carry it to the most remote point in the room. Such a distribution permits the cool air to drop slowly over the entire zone and return to the inlet of the unit below the breathing line and along the floor. The location of doorways, air vents, and heat exposed walls should be carefully observed, as they have a marked effect on the direction of the air flow and on its uniformity of temperature.

With the suspended type of unit, located within the conditioned space, sufficient outlet air velocity should be provided to give adequate induction and mixing with the room air thereby preventing the immediate dropping of the air stream and resulting objectionable cold drafts.

Where the units are located outside the conditioned space, air distribution is more frequently provided through multiple outlets located in ducts from the conditioning unit. The location of these outlets is quite

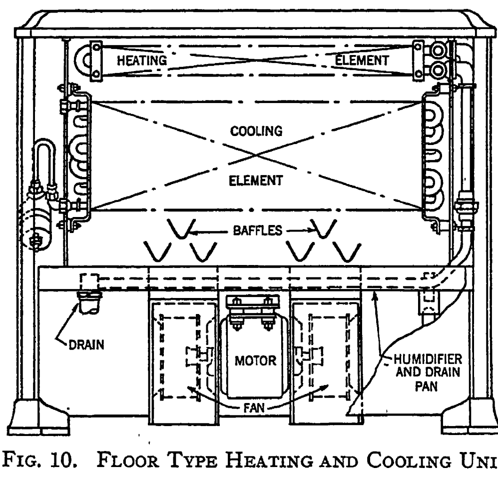

FIG. 10. FLOOR TYPE HEATING AND COOLING UNIT

critical and is influenced both by the building construction, economies of connections and by the distribution of load.

There are a wide variety of outlet types used, and most of these have fixed delivery characteristics, thus requiring careful consideration in their location. Some types of outlets are now available with adjustable vanes thereby permitting some alteration in the delivery of the air stream after installation. This frequently eliminates objectionable down drafts resulting from the impingement of the air stream against posts, pillars, lighting fixtures, and beams.

TYPES OF UNITS

Several types and designs of air conditioning units in production and proposed are available for selection. New designs are constantly appearing, with new improvements, greater capacities, wider range of application and superior construction. It will be impossible to cover in this chapter the many types of construction on the market. Illustrations

of current makes and models will be found in the *Catalog Data Section*. A few typical designs of conditioning units will be described in detail.

An all-year floor type heating and cooling unit for an exposed location and with direct expansion coil supplied with refrigerant from a remotely located compressor is shown in Fig. 10. A cooling coil for use with chilled water may be substituted for the direct expansion coil indicated. The fans below the separate cooling and heating elements deliver the air against deflectors thereby obtaining distribution across the face of the element and preventing condensate from dripping down into the fans. The plate upon which the fans are mounted serves as the drip pan from which the water is conducted to the drain. Separate elements are used for heating and for cooling. Thus this unit may be used automatically for heating and cooling without manual control. When the unit is used for summer conditioning only, the heating coil may be omitted for the

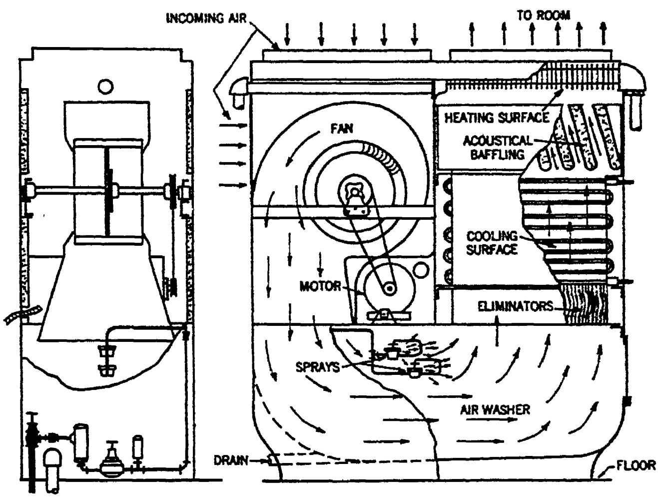

FIG. 11. CONDITIONING UNIT WITH TOP INLET AND OUTLET

installation. The illustration indicates an evaporative type humidifier and drain pan.

Other units are available in which a target spray humidifier is substituted for the evaporative type thereby providing a unit for summer cooling and dehumidification, at the same time supplying humidification in winter for application in rooms with other existing heat sources.

Still another unit is available in which the fans are mounted at the top of the unit delivering directly through a grille and drawing their air supply through the cooling and heating coils. Other variations in proportion and details of construction of this general arrangement are common. With this type of unit, ventilation is usually provided by means of a separate duct connected to the inlet of the unit.

An entirely different arrangement shown in Fig. 11 places both the air inlet and the discharge at the top of the unit. The fan at one side discharges the air downward to the bottom where it turns and passes hori-

zontally through an atomizing spray air washer. The path then continues upward through eliminators, a cooling surface and a heating surface before it leaves the unit. With steam or hot water connected to the heating element, tempered water to the sprays and refrigerated water to the cooling element, this unit gives controlled temperature, humidity, air cleaning, and air movement in both summer and winter. Air washing may be connected in summer, or in intermediate season to remove room

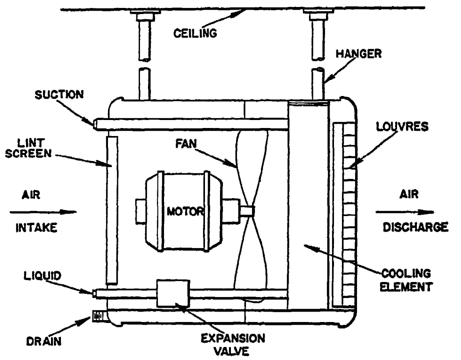

FIG. 12. SUSPENDED PROPELLER FAN TYPE COOLING AIR CONDITIONING UNIT

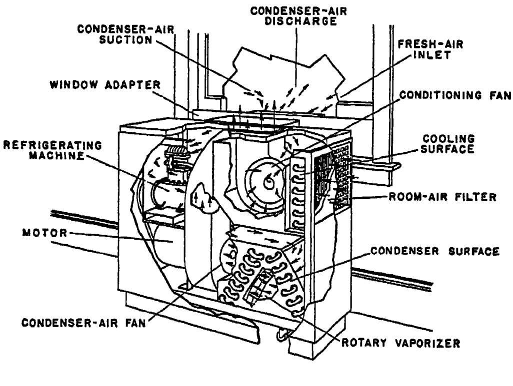

FIG. 13. PORTABLE SELF-CONTAINED CONDITIONING UNIT FOR COOLING

odors. Excess water is run to waste. Acoustical treatment of the housing and outlet baffles permits installation where noise requirements are exacting.

A common type of suspended type unit for exposed location utilizing a propeller type fan and suitable for summer conditioning only is illustrated in Fig. 12. Such units are equipped with either a direct expansion coil or one for chilled water or brine circulation. The outer cabinet is commonly of wood-grained steel or baked enamel and is insulated from the cool air

chamber to prevent external condensation. The drip from the coil is collected in an insulated drip pan and carried to a drain. The inlet to the unit is provided with a lint screen to protect the cooling surface. Such units are normally used for recirculation only but may be connected for ventilation through short full-size ducts. Similar units are available with twin housed fans of the same general construction, although usually such fans draw the air instead of blow it through the coils.

A self-contained completely portable cooling air conditioning unit is illustrated in Fig. 13. For the operation of this unit, it is only necessary that it be located adjacent to a window or shaft to which air connections can be made and to plug in the motors to a convenient light socket. In this unit, the conditioned air enters on the side, passing through a grille, filter, and cooling coil and is delivered vertically to the room through a special motor and fan assembly. Refrigeration is furnished by a recipro-

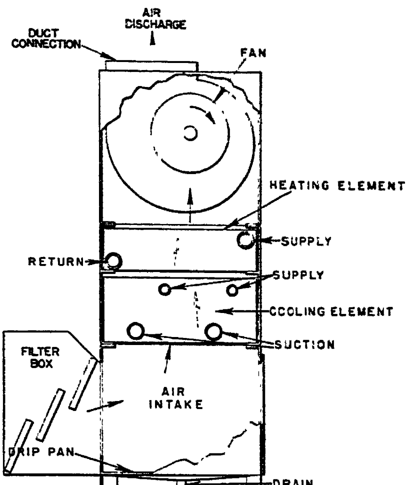

FIG. 14. VERTICAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

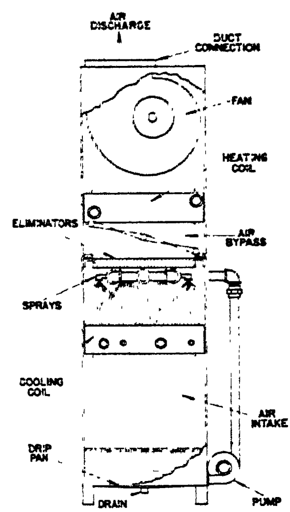

FIG. 15. SPRAY TYPE AIR CONDITIONING UNIT

cating compressor driven from a motor located in the base. This compressor utilizes an air cooled condenser. Air is drawn into the base by a fan mounted on the compressor motor, so arranged that the air passes through the refrigeration condenser and is again discharged out through the window connection. A novel feature of this design is that the condensate from the cooling coil is sprayed over the condenser surface and there vaporized, thus eliminating the need for drain connections. One advantage of this type of conditioning unit is that it may be removed from the occupied space during the winter season when cooling is not needed.

Another type of portable unit for occupied space locations differs from the former in that the compressor is water cooled and a connection to water and drain must be provided in addition to the electric connection. Water and drain lines are carried in a composite hose, especially built for this purpose and connections are usually made to a nearby washbowl.

In order to reduce the starting load, one model has two separate motors brought on to the line at delayed intervals thereby decreasing the initial line surge and reducing light flicker. These water cooled units either eliminate or reduce the need for outdoor air connections. Due to the necessity of water and drain connections they are not as portable as the air cooled type.

Remotely located conditioning units vary widely in details of construction. Figs. 14, 15 and 16 indicate one type built-in sections thereby permitting interchangeability of application with a minimum change in parts. The vertical unit shown in Fig. 14 consists of a fan section, housing one or more fans, mounted on a coil section in which are located a heating coil and a cooling coil, which may be built for either direct expansion refrigerant, chilled water, or brine. These two sections are supported on a third or drip-pan section. The distributing duct system is attached to the fan outlets and returned fresh air connections are made to the drip pan. A filter box is illustrated attached to the drip-pan section. By

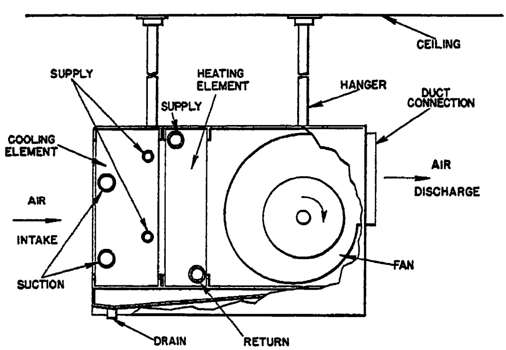

FIG. 16. HORIZONTAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

eliminating the vertical type drip pan and substituting a horizontal drip pan, this unit is converted into a horizontal suspended type conditioning unit for connection to duct work, both with and without filters, as shown in Fig. 16.

A spray type conditioning unit is illustrated in Fig. 15. This spray type unit, which is similar to the arrangement given in Fig. 14, provides for the complete washing of the air and the cooling coil. For winter operation the spray provides means for humidification. The units may also be obtained with by-pass dampers as shown in Fig. 15, to provide control of cooling in summer and humidification in winter. The spray type unit without the cooling coil may be used for humidification and heat control. This type of air conditioning unit is used in industrial process air conditioning as well as for comfort air conditioning.

Residential Central System Units

The previous figures have largely confined themselves to the illustration of all-year-round or summer conditioning units for office or commercial

application. The smaller units have, of course, been applicable to residences. There remains a field of air conditioning units primarily adaptable to residence work. They are in general the outgrowth or adaptation of mechanical warm air systems to conditioning, which are covered in Chapter 20. However, the following illustrations will cover details not included in that Chapter.

In Fig. 17 is shown a conditioning unit which may be operated in conjunction with a hot water or steam boiler. Heat generated in the boiler is supplied to an exchanger which raises the air temperature as it is circulated through the unit. The connection of a cooling coil to a

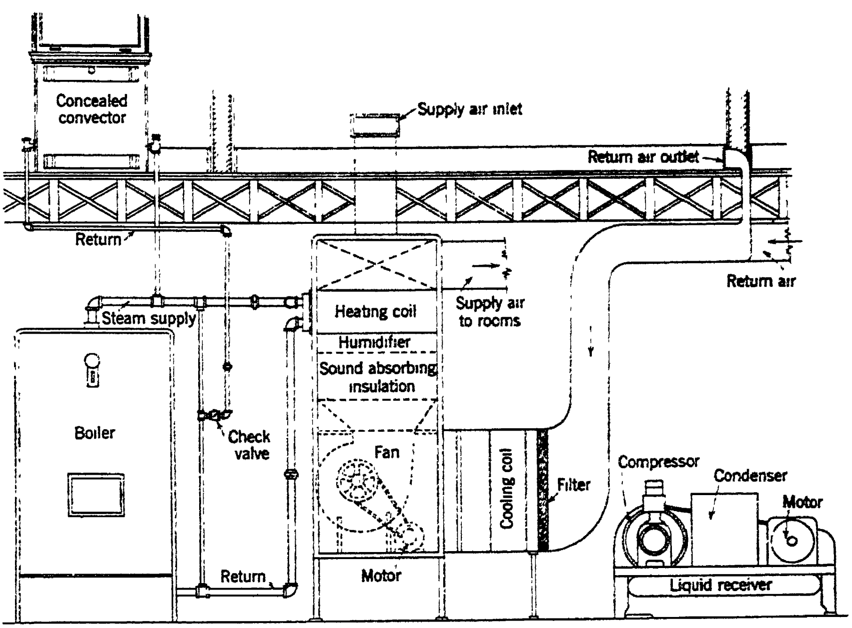

FIG. 17. RESIDENTIAL CONDITIONING UNIT WITH STEAM BOILER

source of refrigeration will provide year-round air conditioning. This type of system is particularly adaptable to a *split* system in which a portion of the residence may be conditioned in the winter and summer while the garage, servants' quarters and less freqently used rooms may be provided with radiator or convector heating directly from the boiler in the winter.

A gas-fired winter conditioning unit is illustrated in Fig. 18 which is equipped with apparatus to filter, heat, humidify and circulate the air in a residence. If a cooling coil is added to the arrangement this unit may also become a year-round conditioner.

A diagram of a direct-fired fuel oil conditioning unit is given in Fig. 19. A circulating fan forces the filtered air over a heat exchanger through which the combustion gases from the oil burner are being directed. A

cooling section may be placed in the air inlet, with cold water, or refrigerant being circulated through the cooling element.

BASIS OF RATING

In the past, the unit air conditioning industry has been handicapped by the lack of any standard method of rating. A proposed code giving a Standard Method of Rating and Testing Air Conditioning Equipment[8] has recently been prepared.

On the basis of this new code, air conditioners are to be classified primarily as *free delivery type* and *pressure type*, where delivery will be measured in cfm standard air at specified fan speed. Pressure type units will specify the delivery against various total fan static pressures. Cooling

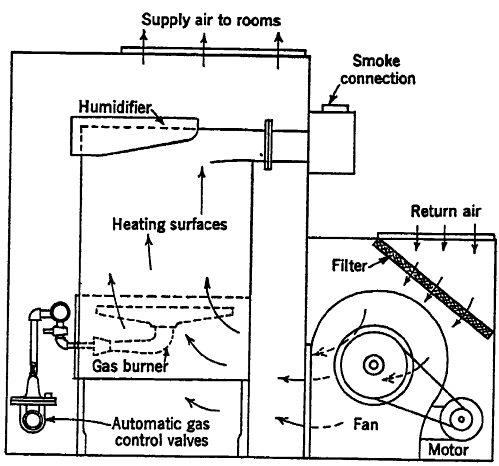

FIG. 18. GAS-FIRED FURNACE CONDITIONING UNIT

capacity will be expressed in total Btu per hour and this total will be subdivided into sensible heat cooling effect and dehumidification or latent heat effect. Cooling capacities will be given on entering air temperatures of 85 F, 50 per cent relative humidity with 40 F refrigerant temperature for comfort conditions and 45 F, 85 per cent relative humidity with 30 F refrigerant temperature for commercial applications.

The duty for heating surfaces will be specified in Btu per hour for 70 F entering air temperature based on 2 lb gage steam pressure or 180 F entering water temperature with 20 F drop. Humidification will be specified in pounds of water evaporated per hour at 70 F and 30 per cent relative humidity entering air conditions. The *Catalog Data Section* gives ratings of current models offered by leading manufacturers.

[8]Loc. Cit. Note 1.

METHODS OF CALCULATING CAPACITIES

The methods of calculating heating and cooling loads for conditioning units are similar to those described under Chapters 7 and 8. Certain manufacturers have adopted simplified and approximate methods which through experience they have found applicable to unitary equipment. Such methods involve certain averaging approximations which are suitable for estimating purposes but many of which require rechecking on a more accurate basis before actual installations are made.

The greatest error in calculating cooling loads is apt to be introduced in the failure to appreciate the magnitude of the latent heat loads and the

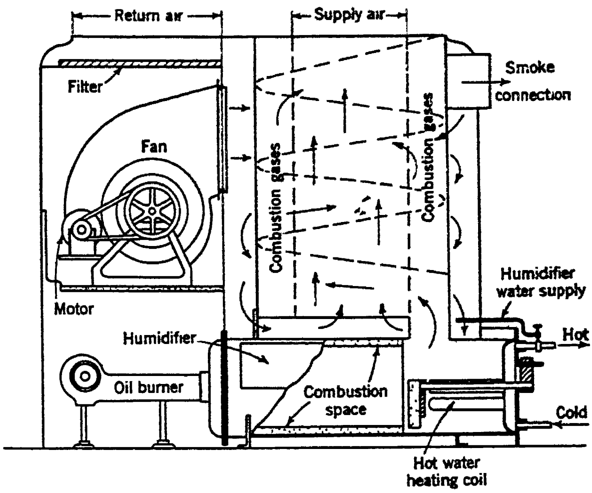

FIG. 19. OIL-FIRED CONDITIONING UNIT

relationship between the latent heat removal and the refrigerant temperature. It is extremely important that the proper balance be obtained between the refrigerant temperature, the conditioning unit surface and the conditions to be maintained within the occupied space. Unsatisfactory conditions often result through the attempt to apply units with a source of refrigeration which gives too high a surface temperature. Such conditions may be obtained when well water of too high a temperature is used or when a direct expansion evaporator is connected to a refrigeration compressor of inadequate size. The high refrigerant temperature even though it may give adequate dry-bulb temperatures, due to over-size conditioning units, will not give proper humidity control due to its inability to furnish sufficient dehumidification.

The use of surface coolers with widely extended fins may lead to similar results since the large ratio of extended surfaces gives an average surface temperature considerably above the refrigerant temperature within the tubes. All these factors must be kept in mind when making the selection

of equipment. It is furthermore vital that the calculation of a cooling load be based on an accurate survey before the selection of the equipment is made.

COSTS

Due to the rapid development of the air conditioning industry and the great progress that is being made each year, it is impossible to give any cost figures that will be of value. There are, however, certain factors that influence the cost of unit air conditioning installations.

1. Since the cost of the total job involves material cost plus installation labor and since through the use of unitary equipment, material costs can be kept to a minimum, every effort should be made to simplify installation.

2. Self-contained units in the small sizes now available, probably represent the lowest cost individual installations. They have, however, their limitations.

3. The floor type all-year-round air conditioning units for the occupied space with a remotely controlled compressor, heating sources being either the existing heat system or steam connections to the unit, probably afford the lowest cost all-year-round service for most individual rooms. This is particularly true in the case of residences. With offices, this will probably be true if the compressor can be located immediately adjacent to the conditioned space, as for example in a closet or nearby storeroom. The expenses increase rapidly as the distance from the unit increases.

4. For multiple rooms or offices, the remotely located unit with connecting ducts probably represents the most economical installation. Such installations are also particularly adaptable to stores, residences and small commercial installations.

Costs of operation vary widely depending entirely upon the cost of power and water. Water costs in the larger installations are being materially reduced through the use of cooling towers and special types of condensers. The normal expense of operating the cooling system is considerably in excess of that of winter heating both as to the first cost and as to operation. Consequently, the more rapid growth of air conditioning has been along commercial lines where it has represented an actual profitable investment resulting in increased business returns rather than along the lines of residential comfort cooling where it still represents a luxury in comfort.

MISCELLANEOUS UNITARY EQUIPMENT

There are a number of units available which were not covered in the previous discussion that accomplish only one or two of the functions of air conditioning.

Attic Fans

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature of a house.

Because the low static pressures involved are usually less than $\frac{1}{8}$ in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and

they should be capable of giving about twenty air changes per hour. The two general types of attic fan installations in common use are:

Open attic fans, in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below. Fresh air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the central grille. It is essential that the roof and the attic walls be free from air leaks.

Boxed in fans, in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Fresh air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

The locations of the fan, the outlet openings, and the grilles should be selected after consideration of the room and attic arrangement in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

The operating routine which will secure best results with an attic fan is an important consideration. A typical routine might require that in the late afternoon when the outdoor temperature begins to fall, the windows on the first floor and the grilles in the ceiling or the attic floor should be opened, and the second story windows should be kept closed. This will place the principal cooling effect in the living rooms. Shortly before bedtime, the first floor windows may be closed and those on the second floor opened, to transfer the cooling effect to the sleeping rooms. A time clock may shut off the fan before waking time, or the fan may be stopped manually at a later hour.

A disadvantage arising from the passing of a great amount of outside air through a house is the dust nuisance, which varies considerably in different locations. Persons suffering from allergic diseases caused by air-borne pollens will have their troubles increased with attic type coolers.

Some typical data on an attic fan installation in an average six-room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installation cost	\$75 to \$400, average \$250
Fan data	9000 cfm average, 280 rpm if belt driven, 570 rpm if direct connected, 500 watts input
Operating period	April 15 to October 15, intermittently as weather conditions demand
Power consumption	500 kwh per year for 8 months' operation

Humidifiers

Humidifying units may be installed as part of an air conditioning unit system, or may be installed individually to furnish additional humidity. Fig. 20 illustrates a humidifying unit for installation in connection with a warm air heating system, and as such it is located at the intake of the furnace. The air passes through a lint filter, then through the fans and finally through an air washer or spray system. Surplus spray is eliminated and the air delivered to the air distribution system. In other cases, similar spray type apparatus is used to deliver humidified air through ducts to openings beneath existing radiators in a steam heated residence.

For other steam heated homes, there is a humidifying unit as illustrated in Fig. 21. This unit is normally placed at some central location on the

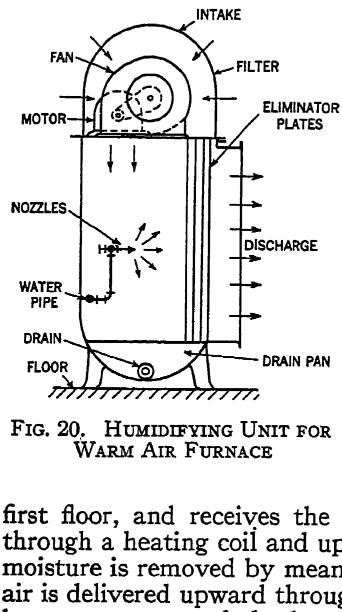

FIG. 20. HUMIDIFYING UNIT FOR WARM AIR FURNACE

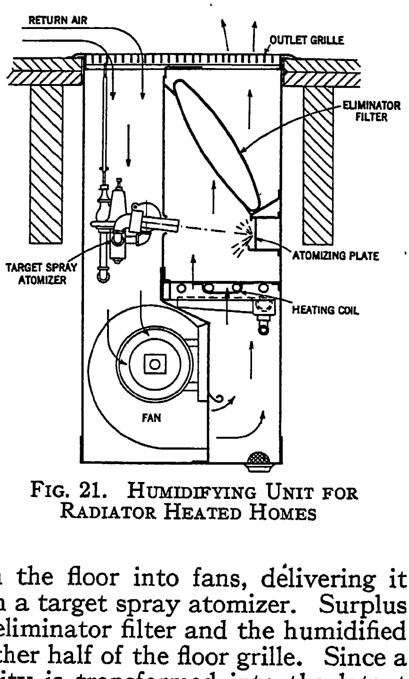

FIG. 21. HUMIDIFYING UNIT FOR RADIATOR HEATED HOMES

first floor, and receives the air from the floor into fans, delivering it through a heating coil and up through a target spray atomizer. Surplus moisture is removed by means of an eliminator filter and the humidified air is delivered upward through the other half of the floor grille. Since a large percentage of the heater capacity is transformed into the latent heat of humidification, this unit does not eliminate any existing steam radiation. It may also be used with hot water systems but its capacity is considerably reduced.

AUTOMATIC CONTROLS

The controls of all unitary equipment represent a vital part of their successful operation. This is particularly true in the case of conditioning equipment where a close inter-relationship exists between the thermostatic room controls and the refrigerating unit controls.

The proper selection of controls and the proper adjustment is extremely important to prevent short cycling of compressors. Furthermore, the proper adjustment of direct expansion valve controls is likewise extremely important. A detailed discussion of control problems is contained in Chapter 37.

PROBLEMS IN PRACTICE

1 ● Distinguish between a unit and a central type of air conditioning system.

In a unit system, the air treating apparatus consists of factory assembled equipment which is shipped substantially complete or in sections and is installed without field fabrication except for the duct connections between the equipment and the point of delivery of the air. Usually the air treating equipment is located closely adjacent to the conditioned space and serves a limited area. A central type of air conditioning system localizes the air treating equipment for the entire area at one point and involves the field assembly of a large number of individual elements. Manufacturer of the unit is responsible for the output and performance of the unit under-rated conditions, whereas, the contractor installing the completely unitary or central type equipment is held responsible for the complete performance of the system.

2 ● Is it satisfactory to use superheated steam in unit heaters?

Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on the saturated steam temperature and not on the total temperature. If unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

3 ● Is it satisfactory to install one unit heater as the total load on a coal fired boiler?

Such an arrangement is impractical if the unit heater is started and stopped in keeping with the room temperature. However, if the room temperature controls the steam pressure and the unit heater is arranged to start when there is steam in the mains and to stop when there is no steam in the mains, such an installation will be satisfactory.

4 ● Will a unit heater with a slow speed fan be more quiet than one with a high speed fan?

Quietness is a function of the type, diameter, blade form, and location of the fan, as well as the speed. For a given fan, slower speeds mean less noise.

5 ● Is it satisfactory to use steam at pressures less than atmospheric for unit heaters or unit ventilators?

If the air inlet temperature is above freezing, steam at any pressure may be used in the heating element of the unit heater or unit ventilator. If the inlet temperature is below freezing the heating element should be filled with steam of at least 5 lb pressure (or with a positive 5 lb pressure differential between supply and return) and the steam supply should never be throttled or the heating element may be frozen.

6 ● In general, what is the primary function of a unit ventilator?

To maintain the desired room air conditions as to temperature, air change, and air cleanliness, without drafts regardless of variations in outdoor temperature, occupancy, sun, heat, and wind.

7 ● What are the usual working parts of a unit ventilator?

The fan and motor assembly, a set of heating elements, outdoor and indoor air dampers, filters (optional), outlet grille, some method of varying the outlet temperature in keeping with the room requirements, and, in the case of some unit ventilators, a method of limiting the outlet temperature to a minimum of 60 F.

8 ● Do all unit ventilators introduce a constant amount of outdoor air?

Certain types employ full recirculation except when outdoor air is obtained by throttling the steam valve on the heating element so the proportion of outdoor air to room air is varied. This is a very economical type of unit ventilator but in some communities it cannot be used because of existing laws which require that some fixed amount of outdoor air be introduced whenever the room is occupied. Certain types of units are designed to always take in a minimum quantity of air from the outside and to automatically vary this with the weather.

9 ● Why are metal surface cooling elements instead of liquid spray chambers used in the design of most air conditioning units and cooling units?

The first cost of the surface cooling type of unit is considerably less than the cost of spray type equipment. Further, the requirements of many industrial air conditioning jobs and of all comfort cooling jobs where unit equipment is applicable can often be effectively met with the use of surface type units, with a reduction in the space required for making the installation. Where space conditions are especially limited, the cross-sectional area of the surface cooler can be reduced because the resulting increase in velocity over the coil surface increases the effectiveness of the surface, whereas an increase in velocity through a liquid spray would reduce its effectiveness.

10 ● Why are air conditioning units with metal cooling surfaces not desirable for all industrial jobs?

Wherever unusually close control of relative humidity is required, a spray type unit will prove to be more satisfactory. Relative humidity control and accurate temperature control, however, can be maintained without difficulty with the use of metal surface units.

11 ● Why is accurate control of relative humidity with surface coolers more or less complicated?

A surface cooler cannot add moisture to the air, and moisture is removed only when the surface temperature is below the entering dew-point temperature. Any change in condition of the entering air will result in a change in the dry-bulb depression of the leaving air. This change in entering condition requires not only a readjustment of the air volume but also a change in the coil temperature, if accurate control over the relative humidity is to be maintained.

12 ● What in general are the characteristics of operation of a unit using surface coils?

For a constant entering dry-bulb temperature and a constant refrigerant temperature any increase in the entering wet-bulb temperature will produce a rise in the leaving dry-bulb temperature with an accompanying reduction in the wet-bulb depression of the leaving air. The sensible heat removed by the unit decreases and the latent heat increases, while the total heat removed also increases. When the dry-bulb temperature of entering air is increased, with constant refrigerant temperature and constant wet-bulb temperature of entering air, the wet-bulb depression of the leaving air increases, and since it is this depression which determines the maintained relative humidity it must be carefully considered when selecting the unit.

Chapter 24

COOLING AND DEHUMIDIFICATION METHODS

Air Cooling Processes, Dehumidification Processes, Practical Combination Methods, Compression Systems, Mechanical Refrigeration, Steam Jet System, Condensers, Evaporators, Refrigerant Pipe Sizes, Operating Methods, Adsorption System, Absorption System, Evaporative Cooling, Reverse Cycle, Ice Systems

COOLING and dehumidifying are closely related in most air conditioning work. Usually a reduction in both temperature and humidity is necessary to produce comfort. Also, it should be borne in mind that: (1) there is a reduction in moisture content whenever air is cooled below its dew-point, and (2) there is a rise in temperature whenever moisture is removed from air by either adsorption or absorption. Consequently, cooling and dehumidification must in most cases be considered together, not as two separate problems, although each can be accomplished separately.

AIR COOLING PROCESSES

In air conditioning either of two arrangements, or a combination of them, is used to accomplish air cooling. The two arrangements are: (a) surface cooling, where the air is passed across a cold metal surface; and (b) spray cooling, where the air is passed through a cold liquid spray, usually water. In either case the surface or the spray liquid must be at a temperature sufficiently low so that heat may be removed from the air. Suitable temperatures for the purpose are obtained by the proper use of

1. Refrigeration; or
2. Water from a cold natural source such as a well, from melting ice, or from application of refrigeration; or
3. Evaporative cooling.

The choice of most suitable source of cooling in any specific case will depend on the accompanying circumstances and can be determined only by a thorough analysis made by a competent engineer.

DEHUMIDIFICATION PROCESSES

Dehumidification may be accomplished in any of three ways, or by a suitable combination of them:

1. By cooling the air below its dew-point temperature thus causing a part of the moisture contained to condense and precipitate.
2. By extracting moisture by adsorption.
3. By extracting moisture by absorption.

As in the case of air cooling, the best dehumidification method can be determined only by a complete analysis taking into account all the circumstances of the particular case being considered. In Chapter 2 the nature of the adsorption and absorption processes are explained and the principal properties of the materials used are presented.

PRACTICAL COMBINATION METHODS

As applied in actual practice these several processes frequently have to be combined in order to produce the desired results. Any or all of the three processes of air cooling listed may be combined with any or all of the three dehumidifying processes to produce both air cooling and dehumidification. One form of combination consists of a multi-stage method whereby moisture is removed from the air and then the resulting mixture is cooled. Stage methods are common where dehumidification is accomplished by the use of adsorbent or absorbent substances. Another method, and one in common use, is to combine the air cooling and dehumidification processes into one step. This is made possible by keeping the temperature of the surface or liquid spray used for cooling below the dew-point temperature of the air to be conditioned. It is the method most commonly associated with comfort air conditioning in current practice. Still another general method consists of what may be called a parallel-flow method wherein the cooling or dehumidification, or both, may be performed by splitting the air stream, performing the process on part of it and then bringing the two parts back together again.

Obviously with so many possible combinations much leeway is left to the designer to determine what shall be done in a practical case. The remainder of this chapter is devoted to a discussion of some of these possible practical methods and the equipment used in applying them. Space does not permit discussing all the great variety possible and only those in reasonably frequent use are included here. Others will occur readily and can be analyzed in a fashion similar to those here treated.

COMPRESSION SYSTEMS

Comfort air conditioning imposes requirements on refrigeration equipment not usually found in general cooling applications so that specially designed apparatus is often required to replace that normally used for industrial cooling. Standard equipment can be adapted to meet air conditioning requirements but extreme care must be taken to determine the limits of its applicability.

In industrial or process cooling systems the load is fairly constant,

noise in operation is not of paramount importance, space is available or relatively cheap, and the cooling system is to a great extent separate or independent of other mechanical equipment. By contrast, air conditioning for space cooling and comfort work in office buildings, theaters and places of public assemblage requires special consideration of all these factors. Space in public buildings is limited, noise interferes with the occupants and the cooling equipment must be adaptable to the other air handling apparatus. Most important, the load fluctuates tremendously and is seasonal.

Types of Compressors

There are many different types of compressors, a number of refrigerants, different types of evaporators, condensers and arrangements of cycle and each type has its particular place in usage. Compressors generally used are of the following types:

1. Reciprocating compressors using a volatile refrigerant.
2. Centrifugal compressors.
   a. Using a volatile refrigerant.
   b. Using water as a refrigerant.
3. Rotary compressors using a volatile refrigerant.
4. Steam jet or vacuum systems using water as a refrigerant.

Reciprocating compressors are generally used with any low pressure refrigerant such as dichlorodifluoromethane, monofluorotrichloromethane, methyl chloride, ammonia and sulphur dioxide. These compressors have been developed to a point where their efficiency is high and their operation very satisfactory. Relatively low speed operation makes them desirable for general use in large installations. Generally they are of two types, vertical and horizontal either single or double acting. The horizontal double-acting compressor is not generally used in air conditioning, except when carbon dioxide is used as a refrigerant in the larger industrial systems. Vertical, single acting, encased crank, reciprocating compressors of the uniflow type with valves in the pistons have proven reliable and are used in capacities from 1 hp to more than 100 hp. At present reciprocating compressors are used with more refrigerants than any other type of compression unit. When carbon dioxide is used as a refrigerant, a reciprocating compressor is required because of the extremely high pressures and the relatively high ratio of compression.

Centrifugal compressors using monofluorotrichloromethane, methylene chloride or water vapor can theoretically be used with any of the other refrigerants, but the resulting loss in efficiency with the higher pressure gases limits the centrifugal compressor to the refrigerants sighted. At the present time centrifugal compressors are limited to air conditioning systems of a minimum of about 50 tons. Centrifugal compressors are usually built in two or more stages where the compression ratio is high and their design follows closely that of any other centrifugal equipment such as is found in general service pumps and fans.

Rotary compressors are expanding in use due to the development of new refrigerants. These units are of four common designs, consisting of rotating elements generally referred to as centrifugal, eccentric, gear and blade types. The rotation of the shafts and blades traps the refrig-

erant vapor between the moving elements and the case and delivers it to the condenser at the required pressure. The rolling together of the impellers as in the case of the gear compressor prevents the return of the refrigerant vapor to the low side of the system.

Steam jet compressors which are particularly adapted to large tonnage installations are simple, compact, have no moving parts and produce practically no vibration. However, they are not economical for water temperatures much below 40 F or where the cost of generating steam is higher than the cost of operation with other prime movers.

MECHANICAL REFRIGERATION

While the mechanical refrigeration systems differ in the methods used for compression of the refrigerant vapor, they are fundamentally similar.

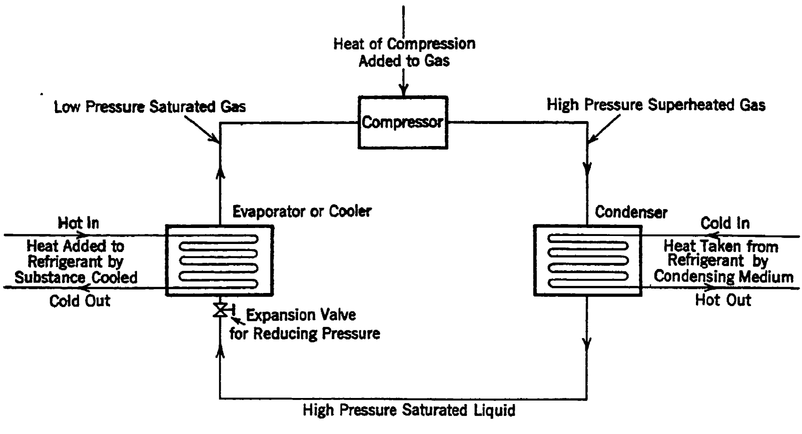

FIG. 1. MECHANICAL REFRIGERATION SYSTEM

Refrigerant vapor usually saturated or slightly superheated, is drawn into the compressor as diagrammed in Fig. 1. It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. In the evaporator the refrigerant absorbs heat from the medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and *pump* it to a higher temperature, where it may be removed by an available cooling medium. In order to conserve refrigerant, virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

Theoretical Mechanical Refrigeration Cycle

The complete mechanical refrigeration cycle may be illustrated on the temperature-entropy diagram, and also on the pressure-volume diagram both of which are shown in Fig. 2.

Considering the theoretical cycle, saturated vapor is drawn into the compressor at *a* and compressed at constant entropy (adiabatically) and then delivered to the condenser at *b*. Condensation occurs at constant temperature T_2 from *b* to *c* with a contraction from the vapor to the liquid volume. The line *cd* represents cooling from the temperature of the condenser to that of the evaporator by an external cooling means. At the same time, the pressure is lowered to P_1. Evaporation then occurs from *d* to *a* at temperature T_1, completing the work cycle *abcda*. Since no external means of cooling the refrigerant liquid is normally available, the

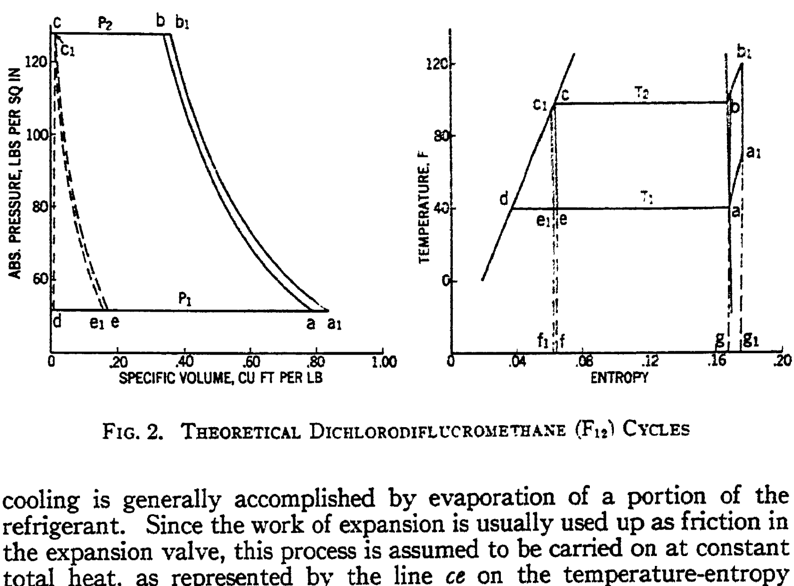

FIG. 2. THEORETICAL DICHLORODIFLUOROMETHANE (F_{12}) CYCLES

cooling is generally accomplished by evaporation of a portion of the refrigerant. Since the work of expansion is usually used up as friction in the expansion valve, this process is assumed to be carried on at constant total heat, as represented by the line *ce* on the temperature-entropy diagram. Thus the refrigerating effect is represented by an area *eagfe*. While the normal theoretical cycle starts with saturated vapor, operation is common at a condition of superheated vapor (as at a_1). Moreover, expansion may start either with a mixture of liquid and vapor or with a sub-cooled liquid, as at c_1, with expansion to e_1. It is obvious that this latter is desirable as it increases the refrigerating effect. Area $a_1b_1cdaa_1$ represents the work of such a superheated cycle, while the area $e_1a_1g_1f_1e_1$ represents the refrigerating effect of the cycle with superheated vapor and sub-cooled refrigerant liquid.

It will be noted on the pressure-volume diagram the volume of the saturated liquid is indicated by a dotted line close to and parallel to the ordinate.

In the discussions in this chapter a slight error is introduced by not including all of the work of pumping the liquid from the low to the high

pressure. This occurs because the liquid line is not a line of equal pressure but of saturation pressures. The error in work per pound of refrigerant figured from total heats, which should be added to the indicated figures is roughly the specific volume of the liquid at the lower pressure and temperature multiplied by the pressure difference in appropriate units. This error may become of some importance in calculations involving carbon dioxide or in problems involving the liquid of any of the refrigerants, as in figuring expansion valve orifices.

Theoretical Work per Pound

The temperature-entropy and pressure-volume diagrams are based on one pound of the refrigerant. Likewise, the theoretical work and the refrigerating effects are conveniently based on a pound of refrigerant. The compression work per pound may be found by several methods.

The temperature-entropy method starts with state point *a*. Since the quality of *a* is known, the heat content of the vapor H_a is known, and also the entropy S_a. Since point *b* lies near the saturation curve it is customary to assume $S_a = S_b$ and with T_2 given, H_b can be determined. If W = work in foot-pounds per pound of refrigerant, then

$$W = (H_b - H_a) \times 778 \tag{1}$$

The pressure-volume method starts with state point *a*, whose pressure and specific volume are known. The work of compression is the adiabatic work of compression from P_1 to P_2, plus the work of expelling the vapor at constant pressure P_2 minus the external work of evaporation of the vapor to volume V_1 at pressure P_1.

$$W = \frac{n}{n-1} \times P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \tag{2}$$

It is frequently helpful to think of the compression of the vapor in terms of head. The head may be likened to a vertical column of vapor in which is located the vapor to be compressed. The compression occurs when the vapor is moved down from a level corresponding to P_1 to a new level corresponding to P_2, in equilibrium with the surrounding vapor. If this process is carried on isentropically, the result will be the same as indicated previously. Then if h is the head in feet,

$$W = h \tag{3}$$

This relationship may easily be seen from the fact that a small difference of head dh divided by the specific volume of the vapor V is equal to the increment of pressure difference dP.

Head is very useful in considering the performance of centrifugal compressors, which merely substitute a centrifugal for the gravity head. It is also useful in considering problems of fluid flow. In these problems, the head per degree can be obtained either by direct calculation or approximately by dividing the total head by the temperature difference $T_2 - T_1$. The velocity head loss can then be calculated in degrees, using the customary formula $V^2 = 2gh$.

Coefficient of Performance

The coefficient of performance of a refrigeration system is the ratio of the refrigerating effect to the work of compression, both expressed in the same units.

The ideal or Carnot coefficient of performance depends upon the temperatures T_1 and T_2 in much the same way as the ideal efficiency of a steam engine depends upon its working temperature, with an inverse relationship.

$$\text{Ideal C. of P.} = \frac{T_1}{T_2 - T_1} \tag{5}$$

Evidently the smaller the compression range, the less power will be required to produce a given refrigerating effect.

The theoretical coefficient of performance of actual refrigerants is always less than the ideal due to the tendency of most refrigerants to superheat when compressed, and due to the heat of the liquid which must be removed. The cycle efficiency is the theoretical C. of P. divided by the ideal for the same temperatures. The cycle efficiency usually changes as the compression temperatures change.

Practical Cycle

Fig. 3 illustrates the pressure-volume and temperature-entropy diagrams for an actual cycle. These diagrams are based upon the compressor receiving vapor superheated and upon sub-cooling of the liquid going to the evaporator. The theoretical cycle is $a_1b_1cc_1e_1a_1$. However, the vapor during compression actually follows line a_1b_2 due to superheating as a result of the inefficient work of compression. The theoretical work of compression is $a_1b_1cda_1$. Added to this is the area $b_2b_1g_1h_1b_2$ on the temperature-entropy diagram which represents the inefficient work of compression (assuming no compressor heat losses). The sum of these areas represents the total work of the compressor per pound of refrigerant, and the ratio of theoretical cycle work to the actual work represents the overall efficiency. It should be noted that area $a_1b_2b_1a_1$ is considered as part

of the inefficient work and is commonly termed the superheat loss. The refrigerating effect per pound is the same for the practical as for the theoretical cycle, working with the same sub-cooling of liquid and superheating of vapor, that is, area $e_1a_1g_1f_1e_1$.

Sources of loss which are usually recognized as reflected by the overall efficiency referring particularly to reciprocating and rotary systems, are as follows:

1. The superheat loss.
2. A pressure loss to and from the cylinder of the compressor. (The line pressure drop between the compressor and the evaporator and condenser, respectively, is usually taken into account separately in the design of the refrigeration system.)
3. Leakage loss through valves and past pistons is quite small in most compressors.
4. With an oil soluble refrigerant, there may be an absorption loss due to absorption and re-evaporation of refrigerant in the oil of the cylinder.
5. Mechanical losses are always present and are usually a large part of the total.

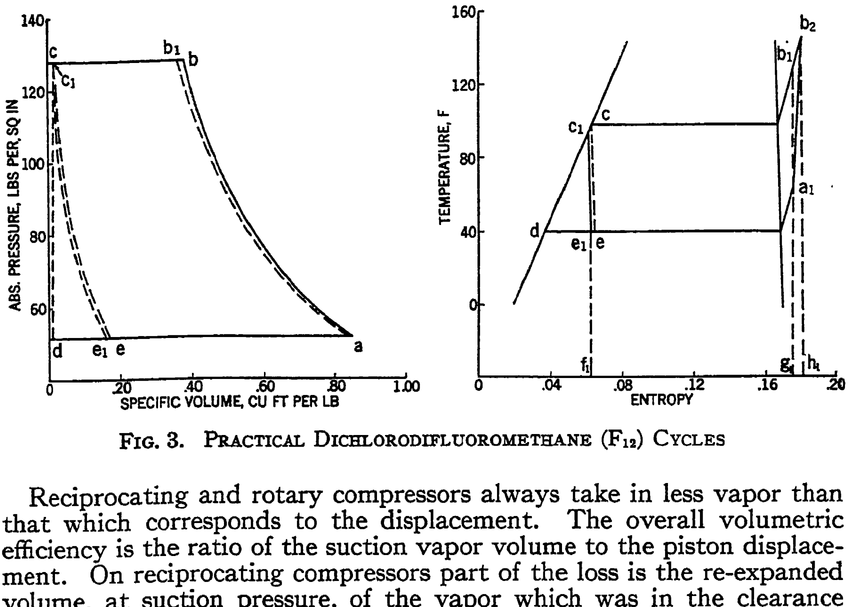

FIG. 3. PRACTICAL DICHLORODIFLUOROMETHANE (F_{12}) CYCLES

Reciprocating and rotary compressors always take in less vapor than that which corresponds to the displacement. The overall volumetric efficiency is the ratio of the suction vapor volume to the piston displacement. On reciprocating compressors part of the loss is the re-expanded volume, at suction pressure, of the vapor which was in the clearance volume. This is expressed by the following equation:

$$\text{Volumetric Efficiency} = 1 - \frac{v_c}{v_d} \times \left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (6)$$

where

v_c = clearance volume.
v_d = cylinder displacement volume.

The balance of the overall volumetric efficiency is known as the superheat volumetric efficiency even though it includes some other sources of capacity loss.

The mechanical efficiency of a reciprocating and rotary compressor must be multiplied by the superheat volumetric efficiency to give the overall efficiency of the compressor.

$$\text{Eff.}_{\text{overall}} = \frac{\text{Vol. Eff.}_{\text{overall}}}{\text{Vol. Eff.}_{\text{reexp.}}} \times \text{Mech. Eff.} = \text{Super. Vol. Eff.} \times \text{Mech. Eff.} \qquad (7)$$

Normally, the volumetric efficiency of a compressor varies with the ratio of compression, while the mechanical efficiency remains virtually fixed. Good standard practices for dichlorodifluoromethane compressors are:

	Low comp. ratio = 2.5 to 1	High comp. ratio = 5 to 1
Vol. Eff.$_{\text{reexp.}}$	94 to 96 per cent	88 to 92 per cent
Vol. Eff.$_{\text{super.}}$	75 to 85 per cent	73 to 77 per cent
Vol. Eff.$_{\text{overall}}$	70 to 81 per cent	64 to 71 per cent
Mech. Eff.	75 to 85 per cent	75 to 85 per cent

These values are for one ton or larger compressors. Part of the difference expresses the change with capacity. With other refrigerants and other types of compressors there will be some further variation.

STEAM JET SYSTEM

The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. The power used for compressing the refrigerant is steam, taken directly from the boiler, thus eliminating the mechanical losses of manufacturing electric current. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiencies of the equipment are somewhat lower than those of the positive mechanical type of compressor; also the condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures, and steam ejectors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 4. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam, and as this requires heat and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point, determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken out of the water circulated, to a somewhat higher absolute pressure, and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture of

entrained air, evaporated water, and impelling steam is discharged into a surface condenser at a pressure which permits the available condensing medium to condense it. The resulting condensate is removed from the condenser by a small pump, from which it can be discharged to the sewer

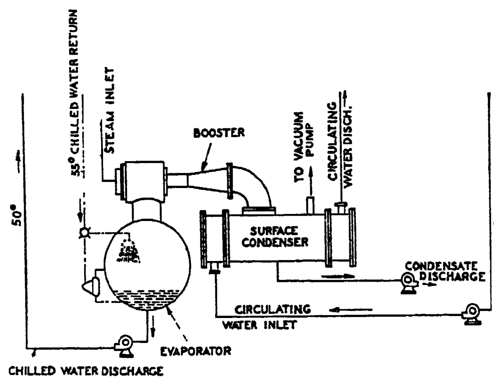

FIG. 4. STEAM EJECTOR COMPRESSION REFRIGERATION SYSTEM

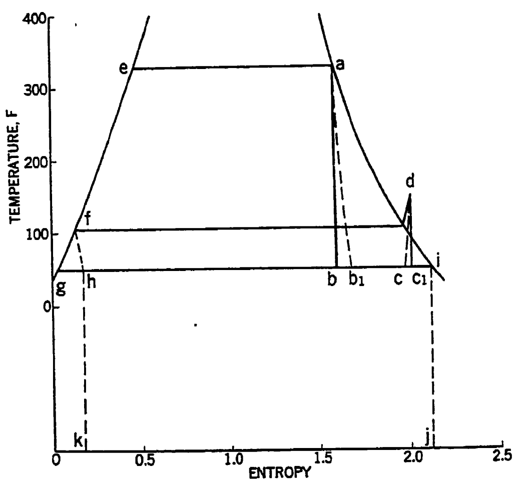

FIG. 5. STEAM EJECTOR TEMPERATURE-ENTROPY DIAGRAM

or returned to the system in the form of make-up water, or part of it may be returned to the boiler feed pump.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A small secondary condenser, of course, is necessary to condense the steam used in the secondary jet.

The performance of the steam ejector may be studied theoretically by the use of the temperature-entropy diagram, Fig. 5. Unlike its usual

application, however, the amount of working fluid is different for one portion of the cycle than for the other. Dry saturated steam under high pressure, for example 100 lb per square inch gage, at a, is expanded through the nozzle of the steam ejector. With 100 per cent efficiency, the expansion would occur along isentropic line ab. Actually, however, most nozzles are only about 90 per cent efficient, the real expansion being along the line ab_1. Since the exact path of the line ab_1 is not known, the work area is normally assumed by using the isentropic giving a work area $abgea$. The velocity at the mouth of the nozzle may be determined in the usual manner using this area and the velocity coefficient of the nozzle.

At the evaporator pressure or slightly below, the vapor from the nozzle mixes with virtually dry saturated vapor from the evaporator. An impact loss also occurs at this point due to the mixture of vapors at different velocities. This results in bringing the state point of the mixture to c. Compression then occurs along the line cd, the work of compression per pound being $cdfgc$. In computing the work area, however, the point c is not actually known. Therefore, the work area c_1dfgc_1 is used in expressing the efficiency of the ejector, the line c_1d being an isentropic. The losses are expressed by nozzle efficiency, impact loss and diffuser efficiency. The work of compression, however, is performed on the mass of the mixture. Thus, the available work is reduced in proportion to:

$$\frac{M_{\text{primary}}}{M_{\text{mixture}}}$$

The impact loss is commonly determined from the formula:

$$MV_{\text{primary}} + MV_{\text{secondary}} = MV_{\text{mixture}} \qquad (8)$$

Common efficiencies for commercial ejectors are: nozzle efficiency 90 per cent, diffuser efficiency 60 to 70 per cent. Customary steam rates in pounds per ton are approximately as follows:

Evaporator temp.	50 F	Steam press. 100 lb per sq in.	Steam press. 12 lb per sq in.
Condenser temp.	105 F	Steam rate 30 lb per hour per ton	Steam rate 45 lb per hour per ton
Evaporator temp.	40 F	Steam press. 100 lb per sq in.	Steam press. 12 lb per sq in.
Condenser temp.	105 F	Steam rate 40 lb per hour per ton	Steam rate 70 lb per hour per ton

CENTRIFUGAL VAPOR VACUUM SYSTEMS

The centrifugal vapor vacuum system functions on the same general principle as the steam jet system, except that a centrifugal evacuator is used to produce the low absolute pressure instead of the velocity of the steam through a jet. Less condenser water is required and a vacuum pump is employed instead of a steam jet purge.

CHARACTERISTICS OF COMPRESSION SYSTEMS

The different types of compression systems have quite different characteristics of capacity and power with varying evaporator temperature and with varying condenser temperature, as will be seen from curves in Figs. 6 and 7.

The capacity of the reciprocating and rotary compressor varies slowly with a change of evaporator temperature, and the variance of power

requirements, in the air conditioning range of operation, is small for a change of evaporator temperature. On the other hand, the capacity and power of the centrifugal machine vary rapidly, and the capacity of the steam ejector also varies considerably. Thus, both these latter types tend to be more nearly self-regulating than the reciprocating and rotary compression type. On the other hand, the operating range of the latter near standard capacity is superior. Although the capacity of the reciprocating and rotary compressor is little affected by the condenser temperature, the power of the compressor is greatly affected, while the reverse is true

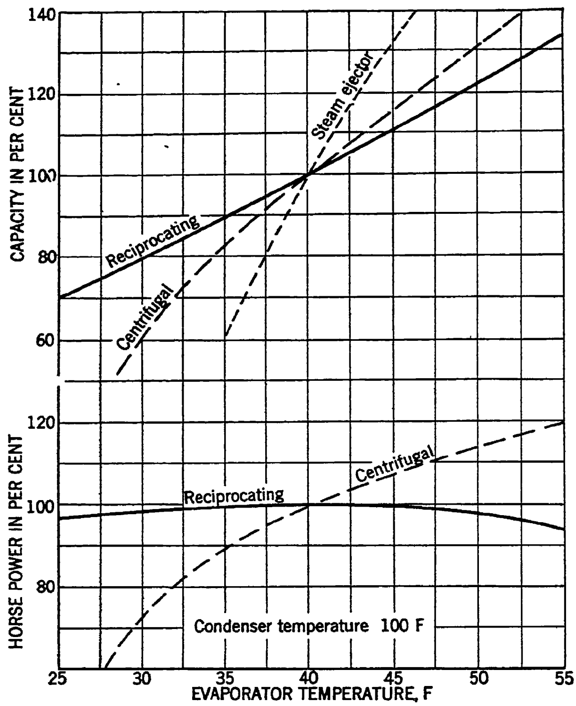

FIG. 6. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

for the centrifugal compressor. As previously indicated, the condenser temperature has no effect on the capacity of the steam ejector type of compressor until a certain point is reached, beyond which the capacity is zero. The steam consumption for the performance characteristic curves shown in Figs. 6 and 7 remains constant for all evaporator and condenser temperatures.

Steam jet refrigeration requires from 3 to 10 times as much condenser water as other types of mechanical refrigeration, but its capacity is not effected by condensing water temperature as long as the water does not greatly exceed 100 F. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where con-

densing water is rather high in temperature. From Fig. 6 it is evident that steam jet refrigeration is better suited for use with evaporator temperatures *above* rather than below 40 F.

CONDENSERS

Condensers used in connection with refrigerating equipment for absorbing the work of compression are of three general designs: (1) air, (2) water, and (3) evaporative.

Air Cooled

Air cooled condensers are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to

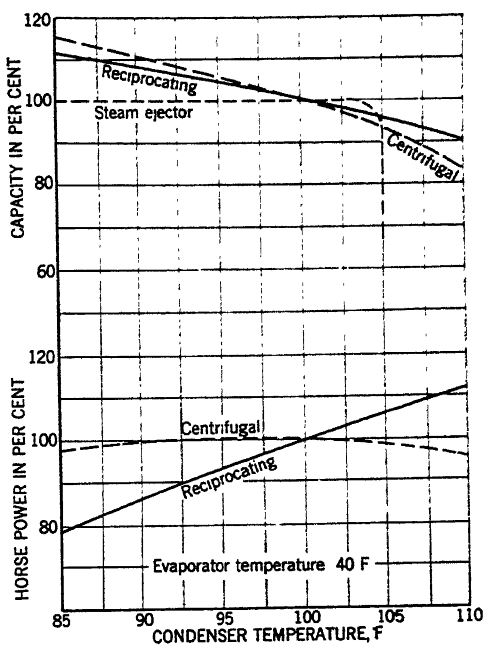

FIG. 7. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

obtain, as for instance in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than can be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot refrigerant gas enters the coil at the top and as it is condensed flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooler air.

The principal disadvantages to air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on warm days requires that equipment of reduced capacity be selected to meet the peak load. Thus, at normal loads the equipment is oversized.

Water Cooled

Water cooled condensers are usually of the double pipe type or the shell and tube type. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes, and refrigeration circulates through the annular space in the outer pipe. Where possible, there should be counter-flow of the refrigerant and the condensing water to maintain maximum temperature differences.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is, therefore, necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet not only the maximum load at normal conditions, but also the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains in that the temperature of the condensing water varies directly with the outdoor temperature and, as pointed out, the refrigeration load also varies with this temperature. Certain economies are possible when a cooling tower is used which cannot be achieved by the use of condensing water from city mains, even where the city water temperature is extremely low. Normally, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of the outdoor temperatures. With the cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water leaving the cooling tower drops considerably, and it has been established that 50 per cent of the time the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is negligible, as the only water required is that used to make up the loss by evaporation in the cooling tower itself. Refer to the section on cooling towers in Chapter 25.

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio, but there are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used, due to fire or other risks especially in densely populated areas, the brine can be cooled in an isolated room or building and then be circulated through the air conditioning equipment in perfect safety because it is used to cool the water or air, without any possibility of direct contact between the air and refrigerant.

REFRIGERANT PIPE SIZES

The selection of proper pipe sizes and frictional pressure losses varies with the installation and the capacity of the system. Generally the suction piping should be selected so that the pressure loss is between 2 and 3 lb per square inch. The pressure drop in liquid lines should be maintained so as to permit no vaporization in the pipes with limiting pressure drops not to exceed 5 lb per square inch. Hot or discharge gas

lines should be limited to approximately 4 lb per square inch pressure drop.

For installations involving piping connections between compressors and evaporative or other remote condensers, pressure drops for discharge or hot gas lines may be referred to in Table 1. Pressure losses in liquid refrigerant lines of various sizes and capacities are given in Table 2. Pressure drops of suction refrigerant pipe lines at varying capacities and refrigerant temperatures may be referred to in Table 3. All tables are for 100 ft of pipe, including an average number of fittings, and for other lengths the losses are proportionate. Allowances should be considered for drops through control and regulating valves which must be added to the other pipe losses to determine the total drop. All copper pipe referred to in these tables are of type *L* wall thickness and are designated by outside diameter.

OPERATING METHODS

There are various methods of designing and operating air conditioning systems to obtain economical operation. Peak outside conditions seldom exist for periods of greater than 3 hr. On many installations there is a peak internal load which may or may not coincide with the peak outside conditions. Thus, each application must be carefully analyzed by the engineer, and the proper equipment installed to satisfy the requirements. Adequate automatic controls should be installed for any system selected.

Where there are a number of small rooms to be conditioned, as for example, a group of hotel bedrooms where the load varies with occupancy and exposure, it may be best to employ individual room units, each with its own control. These individual units may be of the self-contained type (condensing unit, evaporator, fan and controls all in one cabinet) or of the remote type with the condensing units located outside of the room. In some cases it is good practice to use one large condensing unit to serve a group of room evaporator units. Where this is done, the condensing unit must have some type of control which will prevent freezing evaporator temperatures when only a portion of the evaporators are in use. This can be accomplished by means of a back pressure regulating valve which maintains the evaporator pressure at a safe limit, but allows the crank case pressure to fall. Other methods of accomplishing the same result are the use of a variable speed compressor or the use of a partial by-pass from the high side to the low side of the compressor. Any of these three methods of lowering the condensing unit capacity drop the operating cost at the reduced loads, but the operating cost per ton is higher.

Central Distribution Systems

On air conditioning systems using duct distribution, the same general types of control are employed to meet the varying load conditions, i.e. (1) the system may consist of several condensing units and evaporators which are cut in or out, depending upon the demand, or (2) condensing unit capacity may be reduced by using back pressure regulating valves, by-pass valves, or variable speed compressors.

TABLE 1. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE DISCHARGE OR HOT GAS LINES[a]

Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[b]									
	Line Sizes, Inches									
	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝
10,000	2.3	1.0	0.6							
15,000	4.9	2.0	1.0							
20,000	8.5	3.4	1.7	0.6						
25,000		5.3	2.6	0.9						
30,000		7.5	3.6	1.2	0.5					
40,000			6.4	2.1	0.7					
50,000			9.8	3.1	1.0	0.5				
60,000				4.4	1.3	0.7				
70,000				6.0	1.9	0.9				
80,000				8.0	2.5	1.1				
90,000				10.2	3.1	1.4				
100,000					3.8	1.7	0.5			
125,000					6.0	2.6	0.7			
150,000					8.5	3.8	1.0			
175,000					11.6	5.1	1.3			
200,000						6.7	1.7	0.6		
250,000						10.4	2.6	0.9		
300,000							3.7	1.2	0.5	
400,000							6.7	2.2	0.9	
500,000							10.5	3.5	1.5	0.7
600,000								5.0	2.1	1.0
800,000								9.0	3.8	1.8
1,000,000									5.8	2.9
1,250,000									9.5	4.4
1,500,000										6.4
2,000,000										11.3

[a]Soft annealed copper tubing up to and including ¾ in. outside diameter. Hard copper pipe ⅞ in. outside diameter and larger.

[b]Length of tubing includes the average number of fittings.

Another method of providing for economy of operation is to have storage capacity which can be utilized during the peak period. The refrigerating system can be operated for a longer period at maximum efficiency with tanks to store cold water or brine for supplementing the actual output of the refrigerating equipment. However, storage tanks require space and extra apparatus, which increase the cost of the entire system, and further, it is difficult to determine the exact size of the compressor because of the other variables which enter the problem. Depending upon the availability of storage space, the compressor may be designed for any reasonable percentage of the maximum load. On this basis of selection, the smaller the compressor, the larger the storage space, and vice versa.

TABLE 2. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE LIQUID REFRIGERANT LINES

CAPACITY BTU PER HOUR	PRESSURE DROP IN POUNDS PER SQUARE INCH PER 100 FT[a] — PIPE SIZES IN INCHES			
	7/8	1 1/8	1 3/8	1 5/8
100,000	0.6			
125,000	0.9			
150,000	1.3			
175,000	1.8			
200,000	2.3	0.6		
225,000	2.9	0.8		
250,000	3.6	1.0		
275,000	4.3	1.2		
300,000	5.1	1.4		
325,000	5.9	1.6		
350,000	6.9	1.8		
375,000	7.9	2.1		
400,000	9.0	2.3	0.8	
450,000		2.9	1.0	
500,000		3.5	1.3	
550,000		4.3	1.5	0.7
600,000		5.0	1.8	0.8
700,000		6.7	2.4	1.1
800,000		8.7	3.1	1.4
900,000			3.9	1.7
1,000,000			4.7	2.1
1,200,000			6.7	3.0
1,400,000			9.0	4.0
1,600,000				5.1
1,800,000				6.3
2,000,000				7.9
2,200,000				9.2

[a]Length of tubing includes the average number of fittings.

There is a further method of controlling the compressor output which is particularly adaptable to the centrifugal type of machine. This is accomplished by varying the amount of condensing water used with the fluctuation in load demand. Because of the characteristics of the centrifugal type of apparatus, as the condensing water quantity is reduced and the condensing temperature consequently raised, the discharge pressure of the centrifugal machine rises correspondingly and the horsepower input to the machine drops proportionately. While this reduces the total power input to the machine, it does not necessarily reduce the power input per ton of refrigeration developed, as the power input does not drop with a rising discharge pressure as fast as the refrigerating effect is reduced.

TABLE 3. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES

Copper Pipe Actual O.D. Inches	Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
¾	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000		14.5	12.0	9.8	8.3	7.0	5.8
	18,000			15.0	12.3	10.4	8.7	7.2
	20,000				15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000				14.8	12.2	10.2	8.6
	70,000						14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000		14.0	11.0	8.7	7.3	6.2	5.2
	100,000				13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	1.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000		8.4	6.3	4.9	4.0	3.3	2.8
	90,000			8.0	6.2	4.9	4.1	3.5
	100,000			10.0	7.6	6.1	5.0	4.2
	120,000					8.6	7.0	5.9
	140,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

TABLE 3. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (CONTINUED)

Copper Pipe Actual O.D. Inches	Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature, Deg F						
		−10	0	10	20	30	40	50
2 1/8	50,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	100,000	2.6	1.8	1.4	1.1	0.9	0.8	0.7
	150,000	5.6	3.9	3.0	2.4	2.0	1.6	1.4
	200,000	9.8	6.7	5.2	4.1	3.4	2.8	2.4
	250,000	14.8	10.3	8.0	6.3	5.1	4.2	3.6
	300,000		14.5	11.3	9.0	7.2	6.0	5.0
	350,000		19.5	15.3	12.0	9.7	7.8	6.7
	400,000			19.6	15.3	12.5	10.0	8.5
2 5/8	50,000	0.2	0.2	0.1	0.1	0.1	0.1	0.1
	100,000	0.7	0.6	0.5	0.4	0.3	0.2	0.2
	150,000	1.6	1.2	1.0	0.8	0.6	0.5	0.4
	200,000	2.8	2.1	1.7	1.4	1.1	0.9	0.7
	250,000	4.3	3.4	2.6	2.1	1.7	1.3	1.1
	300,000	6.1	4.5	3.7	3.0	2.4	1.9	1.5
	350,000	8.2	6.0	5.0	4.0	3.2	2.5	2.0
	400,000		7.8	6.5	5.1	4.2	3.3	2.7
	450,000			7.7	6.4	5.3	4.0	3.5
	500,000				7.8	6.4	5.0	4.2
	550,000					7.7	6.2	5.1
	600,000						7.4	6.2
3 1/8	200,000	1.2	1.0	0.8	0.6	0.5	0.4	0.4
	300,000	2.6	2.0	1.6	1.3	1.0	0.8	0.7
	400,000	4.5	3.4	2.6	2.1	1.7	1.4	1.3
	500,000	7.3	5.4	4.1	3.3	2.7	2.2	1.9
	600,000		8.1	6.0	4.7	3.8	3.1	2.7
	700,000			8.4	6.5	5.2	4.2	3.5
	800,000				8.6	6.8	5.5	4.6
	900,000					8.7	7.0	5.9
	1,000,000						8.9	7.3
3 5/8	300,000	1.2	0.9	0.7	0.6	0.5	0.4	0.3
	400,000	2.0	1.6	1.3	1.0	0.8	0.7	0.6
	500,000	3.2	2.5	1.9	1.6	1.3	1.0	0.9
	600,000	4.6	3.6	2.8	2.2	1.8	1.5	1.3
	700,000	6.4	4.9	3.8	3.0	2.5	2.0	1.7
	800,000	8.7	6.4	4.9	3.9	3.2	2.5	2.2
	900,000		8.2	6.2	4.9	3.9	3.2	2.7
	1,000,000			7.7	6.1	4.9	4.0	3.3
	1,100,000			9.4	7.3	5.8	4.8	4.0
	1,200,000				8.7	6.9	5.6	4.8
	1,300,000					8.0	6.6	5.6
	1,400,000					9.3	7.6	6.4

[a]Length of tubing includes the average number of fittings.

TABLE 3. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (CONCLUDED)

Copper Pipe Actual O.D. Inches	Capacity BTU per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
4⅛	400,000	1.0	0.8	0.6	0.4	0.4	0.3	0.3
	600,000	2.4	1.8	1.4	1.1	0.9	0.7	0.6
	800,000	4.1	3.1	2.4	2.0	1.6	1.3	1.1
	1,000,000	6.6	4.8	3.7	3.0	2.5	2.0	1.6
	1,200,000	10.0	7.1	5.4	4.4	3.5	2.9	2.4
	1,400,000		10.0	7.5	5.9	4.8	3.9	3.3
	1,600,000			10.0	7.7	6.2	5.1	4.2
	1,800,000				10.0	7.9	6.4	5.3
	2,000,000					9.7	7.9	6.6
	2,200,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

ADSORPTION SYSTEMS

A diagrammatic representation of an open solid material adsorption system is shown in Fig. 9. Two or more beds of the adsorbent are used so that one bed may be used as an adsorber while another is being reactivated. Most adsorption systems use some internal means of heating the adsorbent bed before activation, and cooling it after activation. Thus, the use of relatively high room temperatures and comparatively large amounts of outside air are desirable in connection with these systems. In order to offset the effect of high air temperature, some effort is made to keep the humidity lower than usual.

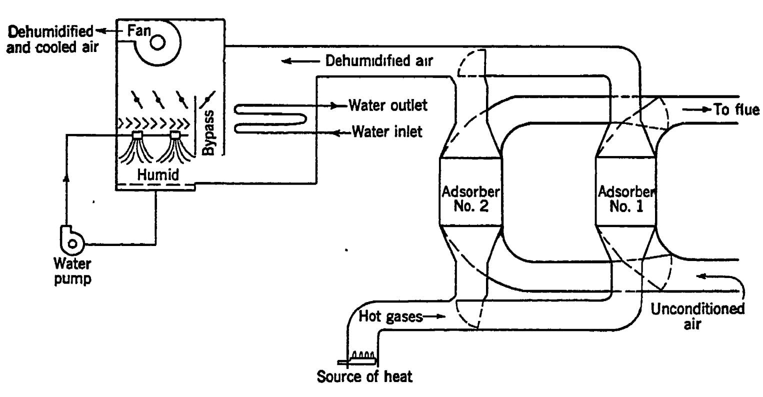

FIG. 9. OPEN SOLID MATERIAL ADSORPTION SYSTEM

Silica Gel

Silica gel has two applications when used to replace refrigeration. In the one principally used, the air from which moisture is to be extracted is taken through silica gel beds by suction or pressure fans, and by means of this process the moisture becomes adsorbed by the silica gel and the air leaves at a lower dew-point and a higher sensible temperature than those at which it entered. If this air is passed over surface coolers in which tap water or another cooling medium is flowing through tubes, a certain amount of sensible heat will be removed. The air leaves the surface cooler or interchanger with the same dew-point with which it emerged from the silica gel beds, but with a lower dry-bulb temperature, although the dry-bulb temperature may be higher than the temperature of the air entering the silica gel beds.

In another method, the first two of the steps outlined are duplicated, and in addition the air is carried through a spray type washer. Because the air enters the washer with a low wet-bulb, and because adiabatic saturation will take place at a temperature close to the entering wet-bulb, considerable cooling of the air can be accomplished; but this can be done only with a consequent increase of the dew-point.

It is necessary to reactivate the silica gel after it has adsorbed about 25 per cent of its own weight in the form of moisture. As reactivation requires a high temperature and since silica gel is only active at low temperatures, cooling of the beds must also be completed before they can be used again. This necessitates three stages in the silica gel containers and requires either three beds of silica gel or one bed divided and automatically put in position. The reactivation is usually done by means of gas or oil fires and the cooling of the beds by means of indirect water cooling or by means of small quantities of dehydrated air taken from the system beyond the interchanger.

Activated Alumina

The application is quite similar to that employed for silica gel; that is, the material is exposed to the air flow and after reaching about 75 per cent saturation is reactivated by removing the moisture adsorbed by means of applied heat. The actual scheme generally followed in the use of this material for continuous service varies somewhat from silica gel inasmuch as the material is placed in three units which are used consecutively for the different steps. These steps permit each unit to operate as follows: (1) in series with the preceding unit, (2) alone, and (3) in series with the following unit. This plan allows for adsorption, reactivation, and cooling, in a manner similar to that used with silica gel.

Taking a single unit, when it is in the (1) step and operating with the preceding unit, the alumina adsorbs approximately 25 per cent of the moisture in the air and takes up about 1.3 per cent of its weight of water. During the second step when it is operating alone, it takes up 100 per cent of the moisture in the air until the weight of the water adsorbed is brought up to about 6.7 per cent. During the third step when the unit is operating with the succeeding unit, it extracts about 75 per cent of the moisture in the air until the water weight adsorbed comes up to about 10 per cent of

the weight of the adsorber. The time allowable for reactivating is equal to the time occupied by the second unit adsorbing alone, plus the time when the second and third units are adsorbing in series, plus the time when the third unit is adsorbing alone, at the expiration of which time the first unit will be again required.

The temperature of air used for alumina reactivation is usually between 300 and 700 F and the air flow rate will have to be higher with the low temperature air than it will be with reactivating air of higher temperature. For example, air at 400 F for reactivating will, at 10 cu ft per hour per pound of alumina, require about 6 hr for reactivation. In the three unit system, after reactivation the cooling of the activated alumina may be carried out with considerable rapidity by using dry air from the adsorption unit for circulation through the unit which has just completed reacti-

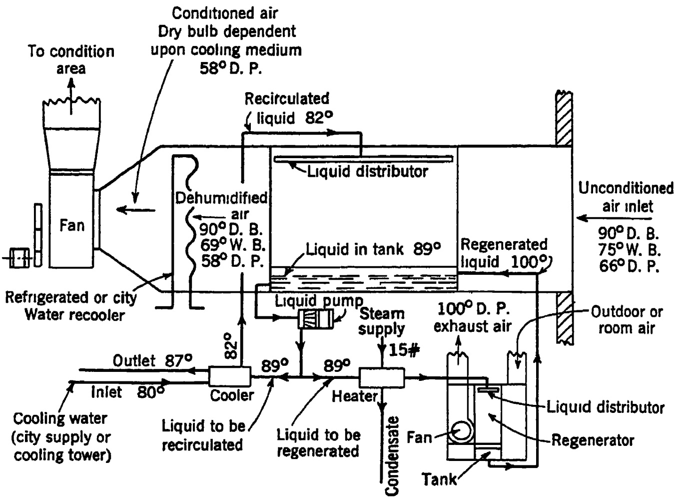

FIG. 10. DIAGRAM OF LITHIUM CHLORIDE ABSORPTION SYSTEM

vation. The final temperature of the unit before it goes back into service should be not over 200 F. As a basis for computing the amount of cooling air required for reactivation, each cubic foot of cooling air has been found capable of removing 2.2 Btu when heated from 85 to 200 F and of providing a sufficient margin of safety in operation.

OPEN ABSORPTION SYSTEMS

The cycle of liquid absorbents are fundamentally the same. A diagram of a lithium chloride absorption system is shown in Fig. 10. The liquid absorbent is brought in contact with air having a certain vapor pressure due to its contained water vapor. The absorbent having a lower vapor pressure, absorbs moisture in the form of water from the water vapor that is in the contacting air. A change of state takes place because there is a rise in temperature in the liquid absorbing the moisture, which is a function of the amount of water vapor condensed from the air stream.

Absorption of moisture by the liquid weakens the concentration of the liquid so that its absorbing capacity is reduced and regeneration, or the driving off of the excess moisture in the liquid, must be performed. A constant density can be maintained by continuously withdrawing a small portion of the total liquid for intensive concentration without varying the vapor pressure of the total mass.

There are two methods of regeneration. One is to boil off the excess moisture by raising the temperature of the solution above the boiling point of the particular concentration. As the salt in the solution does not vaporize, it is not carried off in the boiling process. In the small amounts of liquid diverted to the regenerator for concentration, care should be taken that too much moisture is not driven off, which may cause freezing or solidification of the salts.

The second method of regeneration is to raise the temperature of the solution with ordinary steam coil interchangers to about 225 F. and then passing the solution at this temperature over various types of scrubbers, through which untreated air is circulated. The increase in temperature of the liquid raises its vapor pressure to such an extent that there is an exchange of vapor between the liquid and the air, as well as an equalization of temperature between the air and the liquid absorbent. The air is then capable of taking up part of the moisture from the liquid and carries this excess moisture into the atmosphere with the leaving air.

After this vaporization has taken place, the highly concentrated, hot brine is circulated through an interchanger through which the water used in cooling the main solution can be re-used to reduce the temperature of the concentrated solution to a point where it may be introduced into the main solution tank at only a slightly higher temperature than the main body of the solution.

There are two places in the operation where there is a tendency to raise the temperature of the liquid. One is the absorption of vapor from the air, which changes the latent heat of the vapor absorbed to sensible heat, thus raising the temperature of the liquid and consequently, the temperature of the air. The other is the heat added to the regenerator liquid in order to re-evaporate and carry off the excess moisture which has been condensed in the first stage.

CLOSED ABSORPTION SYSTEMS

The fundamental rule governing the absorption (in a closed system) of a gas by a liquid is Raoult's Law, which states that at any given temperature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same temperature is equal to its *mol* fraction in the solution. The mol fraction, in turn, is equal to the number of mols of substance divided by the total number of mols present. The number of mols in a given weight of a compound is equal to the weight divided by the molecular weight.

This law applies strictly, only to what is known as an ideal solution, that is, one in which the intermolecular forces between the substances present in the solution are equal. Actually, no such solutions exist, so that deviations from Raoult's Law are always found in practice. The

deviation is called positive when the observed pressure is greater than that calculated from Raoult's Law, while the term negative deviation refers to the opposite case. Negative deviations are found wherever chemical attraction exists between the solvent and the solute. Positive deviation occurs when there is a difference in the internal pressure of the components, chemical attraction between them being absent.

In order to make an effective absorption machine, large negative deviations from Raoult's Law must be shown by solutions of the refrigerant in the liquid absorbent, because, the larger the negative deviation, the greater is the amount of refrigerant that can be cycled, using a given weight of absorbent. Cycling a large amount of refrigerant for a given weight of absorbent is important because of the heat required to raise the temperature of the mixture and disassociate the refrigerant and the absorbent. Only the latent heat of the refrigerant can be recovered for useful work.

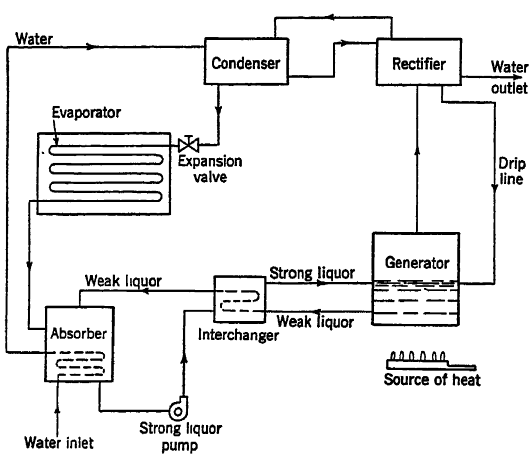

FIG. 11. CLOSED ABSORPTION SYSTEM

Many refrigerant-absorbent combinations have been proposed and quite a number have been tested. Fig. 11 is a diagrammatic representation of a typical closed absorption system. In this system a mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent is condensed. It then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and residual absorbent passes to an absorber where it meets absorbent which is initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor, and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

A cooling medium, ordinarily water, is used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam ejector system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat source is available. Unlike the ejector system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect increases slowly with an increase of evaporator-condenser temperature range.

At the present time the most used refrigerant-absorbent combinations are: (1) water and ammonia and (2) monofluorodichloromethane and dimethyl ether of tetraethylene glycol. With the latter combination the boiling points of the refrigerant and absorbent are sufficiently wide apart that almost pure refrigerant is obtained without the use of a rectifier.

EVAPORATIVE COOLING

Evaporative cooling is accomplished by passing air through a water spray in which the water is being continually recirculated. The air, entering in an unsaturated condition, evaporates a part of the water at the expense of the sensible heat. As this is an adiabatic transfer, the total heat content of the air remains constant, while the dew-point rises and the dry-bulb falls until the air is saturated.

The reduction in dry-bulb temperature is a direct function of the wet-bulb depression of the air entering the spray chamber and the resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air and the efficiency of the spray.

THE REVERSE CYCLE

The idea of heating by the reverse refrigeration cycle has captured the imagination of many people and has been much discussed. In principle, heat is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser. The heat from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical work of compression in relation to the heat delivered is:

$$\frac{T_2}{T_1 - T_2} \qquad (9)$$

where

T_1 = absolute temperature of evaporator.

T_2 = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically and 4 or 5 times practically, as the work put in. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1. Well water is the most desirable since its temperature is high even in the winter and thus a large amount of heat may be removed in relation to the weight of water handled.

2. Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.

3. It has been proposed to obtain heat by freezing water, but this is still in the theoretical stage.

Some of the other factors which act as limitations are the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring extra equipment for a complete heating load, and the relatively high initial cost of equipment at present available for the reverse cycle in comparison with that available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

ICE SYSTEMS

Ice may be used for chilling water for air conditioning work, but its application is limited because of the cost of ice and the difficulty of handling it. While not impossible, the direct cooling of air by ice is rather impractical. The most general method of cooling water with ice is to spray the water over the surface of the ice, insuring as much contact as possible, and approximating the same performance as the Baudelot type of cooler. This cold water is then circulated through cooling coils in the air by means of a pump.

PROBLEMS IN PRACTICE

1 ● Electrically driven dichlorodifluoromethane condensing units are to be used in an air conditioning system, requiring 20 tons refrigerating capacity for conditions of maximum load. An overall analysis of the seasons operating conditions shows an average load factor of 62.5 per cent, and allowing for variable time intervals of operation of refrigeration units installed, three-quarters of the operating season, or 750 hr, would require operation of the equipment at one-half load, and one-quarter of the operating season or 250 hr full load capacity of the refrigeration equipment would be required.

The increased first cost of 2–10 hp, 10 ton condensing units over 1–20 hp, 20 ton condensing unit is, $830.00 installed price, to the customer.

The increased first cost of a 2-speed compressor motor of 20 hp size over a constant speed 20 hp size motor including increased starter cost is $210.00. The efficiency of the 2-speed motor above is 83 per cent at full load speed, and 79 per cent for full load at ½ speed. At ½ speed, full load is ½ total bhp of full load speed.

Discuss the considerations involved in making a decision as to whether a single unit with a 20 hp motor of the 2 speed type would be used in preference to 2–10 hp constant speed units.

The cost of 2–10 hp 10 ton units in excess of 1–20 hp, 20 ton unit with 2-speed moto $830.00 – $210.00 or $620.00, increased first cost. At 15 per cent fixed charges, t represents an increased annual cost of $93.00 for 2 compressors over one compres The advantage of 2 compressors instead of one compressor on an installation of this ty is in the breakdown service provided in the event one compressor is shut down for rep the system could be operated at one-half capacity utilizing the duplicate machine. T motor efficiency of the constant speed unit would be higher at full load than would the efficiency of the 2-speed motor at low speed. Offsetting this latter advant however, is the fact that the condenser on the condensing unit would provide a lo refrigerant condensing temperature for ½ load operation with the same final condens water temperature than would be the case with duplicate units each furnished with own compressor and condenser. Operation at a lower condensing temperature wo provide for a power saving compensating for the lower efficiency of the 2-speed mo when operated at slow speeds. It is, in a case of this kind, purely a question as to whet or not the purchaser would deem an investment of $620.00 more and an increased fi charge of $93.00 a year, advisable to get breakdown service through the installatio duplicate units. In most cases, this increased first cost would not be warranted beca of the fact that satisfactory indoor conditions could not be obtained at full load if o one-half the refrigeration capacity were available.

2 ● For condensing purposes, an air conditioning system uses city water whi has an average 70 F supply temperature. The following table lists the numl of hours per year during which definite wet-bulb temperatures and cor sponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 – 75	100	233
74 – 70	277	183
69 – 65	330	157
64 – 60	277	144
59 – 55	158	79
54 – 50	52	37
	Total 1200 hours	

If the power requirements of a dichlorodifluoromethane refrigeration syste are in accordance with the following data on partial load operation, determi the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Kw per ton	**0.89**	**0.89**	**0.87**	**0.86**	**0.86**	**0.93**	**0.97**

Seasonal power cost:

WET-BULB TEMPERATURE F	TON-HOURS	KWHR
80	6 × 284 = 1,704	1,704 × 0.89 = 1,517
79 – 75	100 × 233 = 23,300	23,300 × 0.89 = 20,750
74 – 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 – 65	330 × 157 = 51,800	51,800 × 0.86 = 44,500
64 – 60	277 × 144 = 39,900	39,900 × 0.86 = 34,300
59 – 55	158 × 79 = 12,500	12,500 × 0.93 = 11,600
54 – 50	52 × 37 = 1,920	1,920 × 0.97 = 1,860
Totals	181,824 ton-hours	158,627 kwh

The 158,627 kwhr at 2 cents per kwhr will cost $3,173.

$$\text{The average consumption will be } \frac{158{,}627 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.873 \text{ kw per ton.}$$

3 ● Using the data from Question 2, if city water costs 20 cents per thousand gallons, and if 1.25 gallons are used per minute per ton, estimate the annual water cost.

$60 \times 1.25 = 75$ gal per ton-hour.

181,824 ton-hours $\times$ 75 = 13,620,000 gal per year.

$$\frac{13{,}620{,}000 \times \$0.20}{1000} = \$2{,}724 \text{ the yearly cooling water cost.}$$

4 ● Using the data of Question 2, if a cooling tower were installed for re-using the condensing water, estimate the annual compressor power cost of a dichloro-difluoromethane refrigeration system if the final temperatures of the water leaving the cooling tower and the kilowatt input per ton are the following:

Tons	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Temperature of water leaving tower, F	**86.7**	**81.8**	**76.5**	**72.1**	**66.4**	**61.3**	**55.6**
Kw input per ton	**1.10**	**0.94**	**0.85**	**0.80**	**0.74**	**0.59**	**0.62**

Wet-Bulb Temperature F	Ton-Hours		Kw per Ton		Kwhr
80	1,704	×	1.10	=	1,875
79 – 75	23,300	×	0.94	=	21,900
74 – 70	50,700	×	0.85	=	43,300
69 – 65	51,800	×	0.80	=	41,400
64 – 60	39,900	×	0.74	=	29,500
59 – 55	12,500	×	0.59	=	7,370
54 – 50	1,920	×	0.62	=	1,200
Totals	181,824 ton-hours				146,545 kwhr

The 146,545 kwhr at 2 cents per kwhr will cost $2,931.

The average consumption will be $\dfrac{146{,}545 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.805$ kw per ton.

5 ● If a steam ejector system were used to secure the refrigeration for the air conditioning system of Question 2, compute the annual steam cost if steam is sold for 53 cents per thousand pounds and if there is an average steam consumption of 20 lb of steam per hour per ton when used with a cooling tower system.

181,824 tons $\times$ 20 lb of steam per ton = 3,636,480 lb of steam.
The 3,636,480 lb at 53 cents per thousand pounds will cost $1,929.

6 ● Discuss the difference in results obtained in cooling and dehumidifying in an air washer from those obtained in a surface cooling coil.

Air leaves a dehumidifying air washer in a saturated condition at a dew point temperature which can be easily maintained at a constant level by controlling the spray water temperature. This saturated air may then be reheated to proper delivery temperature by reheating coils or by mixing with by-passed air.

For a set air velocity and a set mean refrigerant temperature, a given cooling coil is capable of absorbing a definite amount of heat. Whether the air leaving the coil is saturated or not depends then on the entering dry- and wet-bulb temperatures. From, practical operating standpoint, the easiest way to control the output of the cooling coil is by means of the dry-bulb temperature of the conditioned space. This means then that the final dew point will vary somewhat depending on entering air conditions.

Summarizing then, the air washers permit close control over both final dry-bulb and final dew point temperatures, while the surface coolers permit close control over the final dry-bulb only.

Chapter 25

SPRAY EQUIPMENT FOR HUMIDIFICATION AND DEHUMIDIFICATION

Air Washers, Apparatus for Direct Humidification, Spray Generation and Distribution, Self-contained Humidifiers, Atmospheric Water Cooling Equipment, Design Wet-bulb Temperatures, Cooling Ponds, Spray Cooling Towers, Natural Draft Deck Type Towers, Mechanical Draft Towers, Winter Freezing

AIR humidification is effected by the vaporization of water whic always requires heat from some source. This heat may be adde to the water prior to the time vaporization occurs or it may be secure by a transformation of sensible heat of the air being humidified to later heat as the vapor is added to the air. The thermodynamics of th process are discussed in Chapter 1. Dehumidification consists of th removal of moisture from air and may or may not involve the remov of heat from the air-vapor mixture. With spray equipment dehumic ification of air necessitates the removal of heat.

AIR WASHERS

Air washers may be used as either humidifiers or dehumidifiers de pending upon the method of their operation and the temperature of th spray water. The functions of an air washer are to regulate the moistu and heat content of air passing through it and to remove dust and di from the air. As cleaning devices air washers are not as effective as a filters in the removal of dust and dirt.

The construction of commercial air washers is indicated in Figs. and 2. Any air washer consists essentially of a chamber through whic the air passes and comes in intimate contact with water. This chambe may be built of either wood, stone, or sheet metal; the latter being th almost universal material of construction. The lower portion of th washer chamber serves as a sump for the water passing to its botton

Contact between the air and the washer water is secured: (1) b breaking the water into a very fine mist, (2) by passing the air ove surfaces which are continuously wetted by water, or (3) by a combinatio of water sprays and wetted plates. Scrubber-plate types of washers a

used largely to wash heavy reclaimable products from the air, and are generally composed of one to three eliminator-type baffle scrubber plates across the air stream. Water is generally supplied at the tops of the scrubber plates by flooding nozzles placed across the top of the washer. Spray washers have one or more banks of water atomizing nozzles placed in the air stream above the level of the water in the sump. The direction of the water sprays may be against the air stream, with the air stream, or with one bank spraying with the air stream and one bank of nozzles spraying against it. The number of nozzles required depends upon their design, the quantity of air handled, and the arrangement of the nozzles.

Scrubbers generally consist of eliminator-type baffle plates placed in the air stream to cause several reversals of the direction of air flow. The scrubber plates are more effective as air cleaners than as humid-

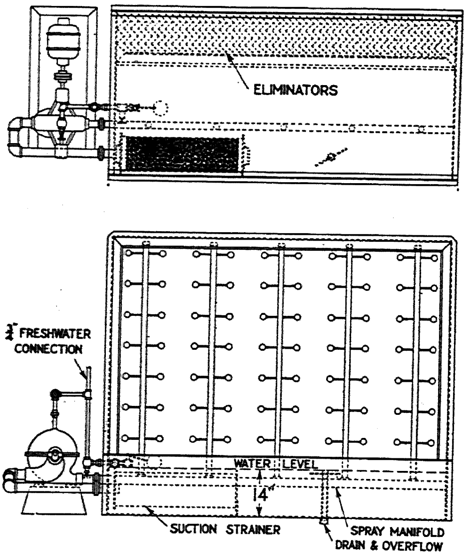

FIG. 1. TYPICAL SINGLE BANK AIR WASHER

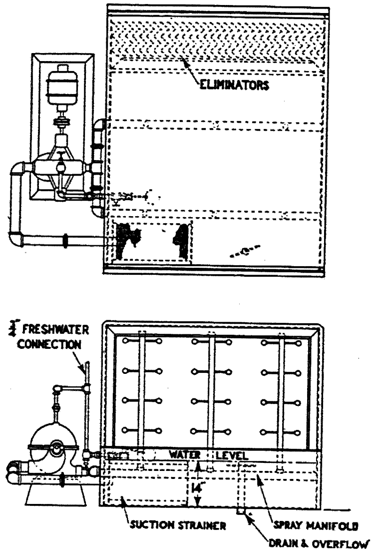

FIG. 2. TYPICAL TWO BANK AIR WASHER

ifiers. All washer chambers should have inlet diffuser plates to aid in producing more uniform velocities of air flow through the washer spray chamber. These inlet vanes also aid in preventing spray water from being thrown into the air duct ahead of the washer. At the outlet end of the washer suitable flooded eliminator plates, which will cause from 4 to 6 reversals of the direction of air flow, should be installed for the purpose of removing drops of unvaporized water from the leaving air. When the air carries sulphur and gases mixed with it the spray water may become acidulated and special consideration must be given to the selection of eliminator plates to reduce the corrosive action.

Essential items in air washer operation are: uniform distribution of the air across the chamber section above the level of the water in the

sump; moderate velocities of air flow, 300 to 600 fpm in the spray chamber; an adequate amount of spray water broken up into a fine mist throughout the air stream; sufficient length of air travel through the water spray and over thoroughly wetted surfaces, and the elimination of free moisture from the air as it leaves the unit.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the overall efficiency of heat transfer between the air and the heating or cooling medium. A multi-stage washer is equivalent to a number of washers in a series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 fpm through the gross cross-sectional area of the unit above the tank. At this rating spray type washers handle about 2½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed.

For a single stage air washer, a 15 F drop in wet-bulb temperature of the air passing through the washer is about the maximum that should be anticipated. For greater decrease in wet-bulb temperature, multi-stage washers should be utilized. A rise of 6 F should be the calculated maximum for the spray water.

The area of a washer may be dictated by space limitations outside the washer, such as headroom, or by the inside space requirements, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays, (the *leaving* eliminators require about 1 ft 6 in., *entering* eliminators about 1 ft).

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

HUMIDIFICATION WITH AIR WASHER

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air, (2) preheating the air and washing it with recirculated spray water, and (3) using heated spray water. In any problem of air washing the air should not enter the washer with a dry-bulb temperature less than 35 F so that there will be no danger of freezing the spray water.

When method 1 is used the principles of adiabatic saturation described in Chapter 1 are involved. The process is one of evaporative cooling as the dry-bulb temperature of the air is reduced and the total heat of the air and water-vapor mixture is unchanged. Moisture is added

to the air and a part of the sensible heat of the initial mixture is transformed to latent heat as evaporation of some of the spray water takes place. Theoretically the spray water and the dry- and wet-bulb temperatures of the air should come to the wet-bulb temperature of the air entering the washer and the air should leave the washer adiabatically saturated at the entering wet-bulb temperature. Due to limitations of air washer construction and operation air is not generally completely adiabatically saturated. This introduces an item into the calculations which is known as humidifying or saturating efficiency. This efficiency is the ratio of the actual reduction of dry-bulb temperature to the reduction of dry-bulb temperature theoretically possible. Expressed as a percentage humidifying efficiency is:

$$e_h = \frac{(t_1 - t_2)\,100}{t_1 - t^1} \tag{1}$$

where

e_h = humidifying efficiency, per cent.
t_1 = initial dry-bulb temperature, degrees Fahrenheit.
t_2 = final dry-bulb temperature, degrees Fahrenheit.
t^1 = initial wet-bulb temperature of the entering air, degrees Fahrenheit.

The humidifying or saturating efficiency of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocities of air flow are more conducive to higher humidifying efficiencies than high velocities of air flow. The following may be taken as representative humidifying or saturating efficiencies of air washers for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The air leaving the washer may require the use of a reheater coil to produce the required dry-bulb temperature and relative humidity.

When air of a given specific humidity has a low initial dry-bulb temperature it may be preheated before it enters a washer using recirculated spray water. The preheating of the air increases both the dry- and wet-bulb temperatures and lowers the relative humidity, but not the specific humidity of the air. With an increased wet-bulb temperature, the air is capable of accumulating more moisture by the process of adiabatic saturation and the final specific humidity and the final dry-bulb temperature of the air as it leaves the washer will be higher. An addition of sensible heat by the preheater takes place prior to the air entry into the washer. In the case of method 2 the process of humidification within the washer is similar to method 1. The final desired conditions are secured by adjusting the wet-bulb temperature of the entering air and the use of a reheater when such is necessary.

Method 3 involves heating the spray water to a temperature equal

to the air and a part of the sensible heat of the initial mixture is transformed to latent heat as evaporation of some of the spray water takes place. Theoretically the spray water and the dry- and wet-bulb temperatures of the air should come to the wet-bulb temperature of the air entering the washer and the air should leave the washer adiabatically saturated at the entering wet-bulb temperature. Due to limitations of air washer construction and operation air is not generally completely adiabatically saturated. This introduces an item into the calculations which is known as humidifying or saturating efficiency. This efficiency is the ratio of the actual reduction of dry-bulb temperature to the reduction of dry-bulb temperature theoretically possible. Expressed as a percentage humidifying efficiency is:

$$e_h = \frac{(t_1 - t_2)\,100}{t_1 - t'} \qquad (1)$$

where

e_h = humidifying efficiency, per cent.
t_1 = initial dry-bulb temperature, degrees Fahrenheit.
t_2 = final dry-bulb temperature, degrees Fahrenheit.
t' = initial wet-bulb temperature of the entering air, degrees Fahrenheit.

The humidifying or saturating efficiency of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocities of air flow are more conducive to higher humidifying efficiencies than high velocities of air flow. The following may be taken as representative humidifying or saturating efficiencies of air washers for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The air leaving the washer may require the use of a reheater coil to produce the required dry-bulb temperature and relative humidity.

When air of a given specific humidity has a low initial dry-bulb temperature it may be preheated before it enters a washer using recirculated spray water. The preheating of the air increases both the dry- and wet-bulb temperatures and lowers the relative humidity, but not the specific humidity of the air. With an increased wet-bulb temperature, the air is capable of accumulating more moisture by the process of adiabatic saturation and the final specific humidity and the final dry-bulb temperature of the air as it leaves the washer will be higher. An addition of sensible heat by the preheater takes place prior to the air entry into the washer. In the case of method 2 the process of humidification within the washer is similar to method 1. The final desired conditions are secured by adjusting the wet-bulb temperature of the entering air and the use of a reheater when such is necessary.

Method 3 involves heating the spray water to a temperature equal

As in the cases of humidification by use of an air washer the heat necessary for the vaporization of the moisture added to the air is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the total heat of the air remains constant but the dry-bulb temperature of the air humidified is reduced.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray which cannot be permitted when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located

pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled to the wet-bulb temperature. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is quickly evaporated and the resulting vapor is rapidly and thoroughly diffused. This effective distribution of fine spray over the maximum possible area insures complete and extremely rapid vaporization even at the highest humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

The spray and high-duty types of humidifiers have many features in common but the latter, because of its finer spray and greater capacity, is often considered better adapted for producing high humidities.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

Where large quantities of power are generated in a limited space and where a comparatively high relative humidity is required, it is often feasible and economical to use a combination of direct and indirect humidification. The indirect humidification provides the desired quantity of ventilation and cooling, and the additional direct humidification provides for increase in humidity without interfering with the ventilation or the cooling effected by the indirect system.

In general, it may be stated that direct humidification is most satisfactory where high humidities are desired but where little cooling, ventilation or air motion is required. Therefore, the indirect system is most applicable where either low or high relative humidities are desired with maximum cooling and ventilation effect. For conditions that require an unusually large amount of heat to be absorbed by ventilation, together with the maintenance of high humidities, it is often preferable to make use of the combination system of indirect and direct humidification. If the indirect system alone were used it would mean an unusually large volume of air to be handled, which might interfere, due to air motion, with production, even though it would result in greater cooling effect. If direct humidification alone were used, no ventilation would be obtained, with consequently higher room temperatures.

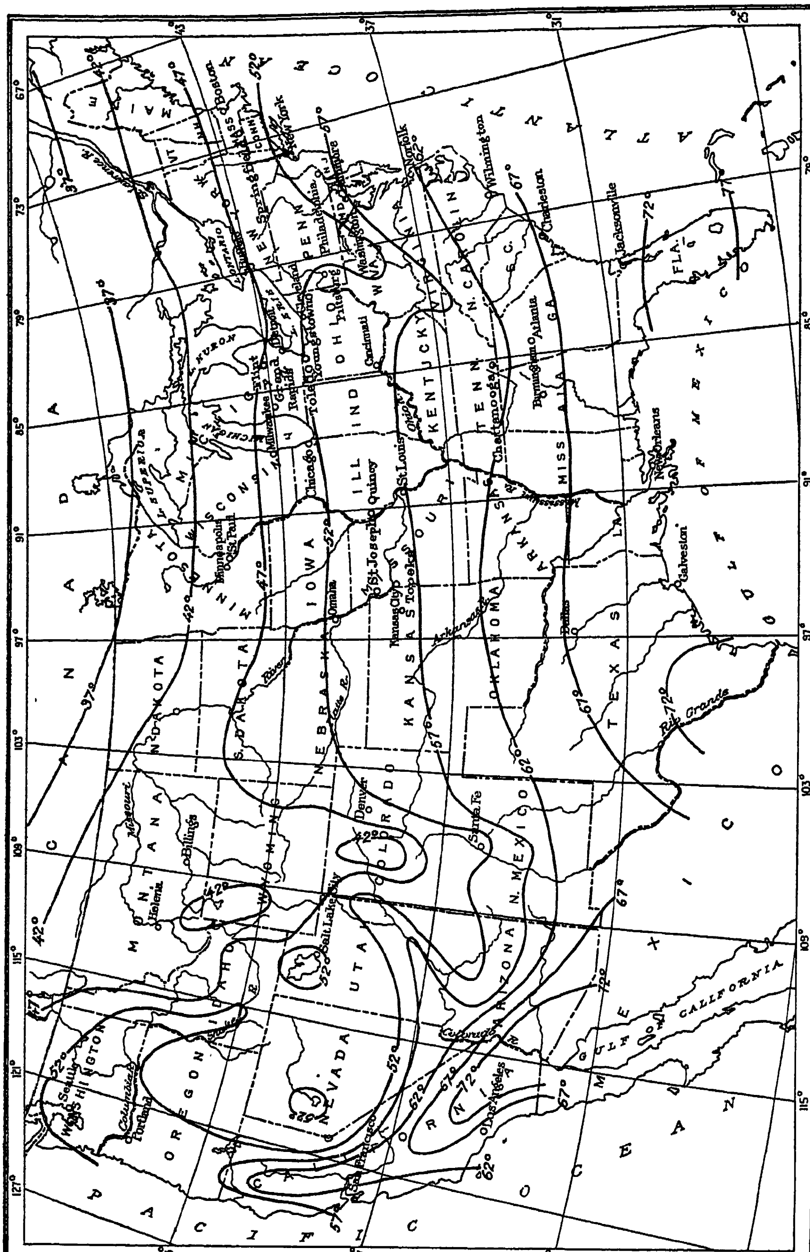

FIG. 4. APPROXIMATE WELL WATER TEMPERATURES AT DEPTHS OF 30 TO 60 FT

AIR DEHUMIDIFICATION WITH WASHERS

Moisture removal from an air-vapor mixture can be accomplished by use of an air washer so long as the temperature of the spray medium is less than the dew-point of the air passing through the unit. The final dry-bulb temperature and the percentage of the saturation of the air leaving a dehumidifier washer are dependent upon: the air velocity, the length of air travel through the sprays, the dry- and wet-bulb temperatures of the entering air, the spray temperature, the number of spray banks and nozzles, the quantity of spray medium handled, and the effectiveness of the nozzles in breaking the spray into a fine mist.

Both sensible and latent heat are removed in the process of dehumidification by cooling. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat removal takes place as condensation occurs. Therefore, the lower the spray temperature the greater the amount of moisture removal per pound of dry air all other conditions remaining the same. The spray temperature should be controlled to 1 or 2 F below the desired leaving dew-point temperature of the air. Washers with two or more banks of sprays are usually selected for comfort air conditioning installations. Such washers will cool the air to within 1 or 2 F of the spray temperature.

Where a limited supply of cold water is available multiple stage washers may be used to an advantage. The cool water is pumped through the multiple spray systems in series. By this arrangement the entering air is cooled first by the warmer water and finally by the cooler water which gives the maximum amount of cooling with the minimum amount of water. The approximate temperatures of water from non-thermal wells at depths of 30 to 60 ft are given in Fig. 4[1]. Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in this country and Canada.

Air washers using refrigerated spray media generally have their own recirculating pumps. These pumps deliver to the washer sprays a mixture of water from the washer sump, which has not been re-cooled, and refrigerated water. The quantities of each of the portions of the spray medium are controlled by a three-way or mixing valve actuated by a dew-point thermostat located in the washer air outlet.

An illustration of a cooling and dehumidifying calculation is given in Example 3 of Chapter 1.

ATMOSPHERIC WATER COOLING EQUIPMENT

In the operation of a refrigerating plant or a condensing turbine, one of the main problems is the removal and dissipation of heat from the compressed refrigerant or the discharged steam. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger, from which water it may then be dissipated in a number of ways. If the plant is situated on the banks of a river or lake, an intake

[1]Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S. Geological Survey*, Water Supply Paper No. 520 F).

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES[a]

STATE	CITY	TEMP. F
Ala.	Burmingham	84
	Mobile	73
Ariz.	Phoenix	81
	Tucson	80
Calif.	Anaheim	60
	Berkeley	69
	Fresno	72
	Fullerton	75
	Glendale	68
	Los Angeles	75
	Oakland	69
	Ontario	70
	Pasadena	82
	Pomona	75
	Riverside	78
	Sacramento	72
	San Bernardino	65
	San Diego	82
	San Francisco	62
	Whittier	75
Colo.	Denver	75
Conn.	Bridgeport	66
	Hartford	73
	New Haven	76
	Waterbury	72
D. C.	Washington	84
Del.	Wilmington	83
Fla.	Jacksonville	80
	Miami	80
	Tampa	77
Ga.	Atlanta	87
	Macon	80
Ill.	Chicago	76
	Cicero	76
	Evanston	73
	Peoria	67
	Rockford	59
	Springfield	82
Ind.	Evansville	86
	Gary	75
	Indianapolis	80
	South Bend	61
	Terre Haute	82
Iowa	Cedar Rapids	78
	Des Moines	77
	Sioux City	62
Kans.	Concordia	57
	Kansas City	86
	Topeka	88
	Wichita	72
Ky.	Louisville	85
La.	Baton Rouge	85
	New Orleans	85
Me.	Augusta	60
Md.	Baltimore	67
Mass.	Boston	80
	Cambridge	70
	Fall River	76
	Lowell	50
	Lynn	68
	New Bedford	70
	Salem	68
	Worcester	76
Mich.	Detroit	77
	Flint	70
	Grand Rapids	84
	Highland Park	77
	Jackson	56
	Kalamazoo	53
	Lansing	64
	Saginaw	82
Minn.	Duluth	55
	Minneapolis	80
	St. Paul	77
Mo.	Jefferson City	82
	Kansas City	84
	St. Joseph	84
	St. Louis	85
	Springfield	70
Nebr.	Lincoln	87
	Omaha	87
Nev.	Reno	61
N. H.	Manchester	76
N. J.	Jersey City	63
	Newark	74
	Paterson	78
	Trenton	79
N. Y.	Albany	68
	Buffalo	75
	Jamaica	56
	Mt. Vernon	74
	New Rochelle	75
	New York	72
	Rochester	70
	Schenectady	60
	Syracuse	74
	Utica	69
	Yonkers	70
N. C.	Asheville	74
	Charlotte	85
	Winston-Salem	82
N. M.	Albuquerque	65
Ohio	Akron	76
	Canton	50
	Cincinnati	84
	Cleveland	74
	Columbus	82
	Dayton	60
	Lakewood	82
	Springfield	72
	Toledo	83

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURE[a] (CONTINUED)

STATE	CITY	TEMP. F	STATE	CITY	TEMP. F
Okla.	Oklahoma City	82	Utah	Logan	44
	Tulsa	85		Salt Lake City	60
Oreg.	Eugene	60	Va.	Fredericksburg	75
	Portland	64		Lynchburg	73
Pa.	Altoona	74		Norfolk	80
	Erie	75	Wash.	Olympia	58
	Johnstown	74		Seattle	62
	McKeesport	82		Spokane	51
	Philadelphia	83		Tacoma	57
	Pittsburgh	67	W. Va.	Charleston	85
R. I.	Providence	68		Huntington	78
S. C.	Charleston	80		Wheeling	78
	Greenville	81	Wis.	LaCrosse	54
	Spartanburg	78		Madison	58
S. Dak.	Rapid City	55		Milwaukee	70
Tenn.	Chattanooga	84		Racine	68
	Knoxville	89			
	Memphis	70			
	Nashville	90			
Texas	Amarillo	65	PROVINCE		
	Austin	90			
	Beaumont	86			
	Dallas	86	Alta.	Calgary	64
	Fort Worth	84	B. C.	Vancouver	60
	Galveston	90	Ont.	London	50
	Houston	84		Toronto	63
	Port Arthur	83	P. E. I.	Charlottetown	48
	San Antonio	76	Que.	Montreal	78
	Wichita Falls	85		Quebec	68

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

may be taken upstream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of cooling water is a city supply or a well, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains dissolved salts which would form scale on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1.) Because it is impractical to leave the air in contact

with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess quantity of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of

the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

TABLE 2. CONDENSER DESIGN DATA

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser Deg F	Leaving Hot Water Temperature Deg F	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodifluoromethane	117 lb gage head pressure	100.0	96	93

The temperature range, once the hot water temperature is approximately known, depends upon:

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.
2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September, inclusive) when the wet-bulb temperature was 68 F or above. As these

975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City), with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 8, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

Equipment	Cooling Efficiency—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	45 to 55	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 3 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton.
Condenser turbine	950 to 980 Btu per pound of steam.
Steam jet refrigerating apparatus	1030 to 1150 Btu per pound of steam.
Diesel engine	2800 to 4500 Btu per horsepower.

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles, to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 to 7 ft above the edge of the basin, to supply from 5 to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground if they have an earthen or a concrete basin, or they may be placed on roofs having special waterproof roofing. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a

corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae growths, during warm weather, in cooling towers and spray ponds may be eliminated while the plant is in operation by the use of potassium permanganate. This chemical can be dissolved at the rate of 1 lb in 1¼ to 1½ gal of hot water. About 10 parts of permanganate should be used per million parts of cooling water.

The permanganate attacks the algae, forms a brown covering over it, and causes it to settle. Enough of the permanganate solution should be added periodically to cause the water to have a pink color for a period of from 15 to 20 min. Small additions of the permanganate daily do not give concentrations which are effective. The best results are obtained when sufficient quantities are added periodically at intervals of several weeks, the time intervals being dependent upon local operating conditions. The chemical is non-poisonous and is non-corrosive when used as directed.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 to 80 ft high and from 6 to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1, Chapter 8, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case for a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and

this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 to 600 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside. Such apparatus does not have the counterflow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

TABLE 4. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per square foot	x	x	1	3	4	2
Independence of wind velocity	6	3	4	5	1–2	1–2
Drift nuisance	1	6	5	4	2–3	2–3
Make-up water required	1	6	5	4	2–3	2–3
Pumping head	1	2	3	4–5	4–5	6
Maintenance	2	1	3	4	5	6
Suitability for congested districts	x	5	4	3	1	2
Water quantity required for definite result	6	5	4	1–2	1–2	3

xNot comparable.

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 F range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to avoid excessive deposits in the condensers. An additional amount of make-up water must be added to replace windage, or *drift loss*. This additional amount of water varies from 0.1 to 3 per cent of the quantity of water being circulated, this percentage depending upon the type of equipment and the wind velocity.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 4.

PROBLEMS IN PRACTICE

1 ● What performance tests should be given air washers?

a. Capacity, *b.* Resistance, *c.* Visible entrainment of free moisture, and *d.* Humidifying or dehumidifying efficiency.

2 ● What are different types of air washers?

a. Spray, *b.* Wet scrubber, and *c.* Combination spray and scrubber.

3 ● Upon what air velocity are air washers usually rated?

500 fpm through the area above the tank.

4 ● What is the difference between direct and indirect humidification?

Direct humidification signifies that the humidifiers are within the space to be humidified with distribution produced by the number of humidifiers. With direct humidification there is relatively little air movement.

Indirect humidification signifies that the air is drawn from the enclosure and passed through the humidifier (air washer) and distributed by means of a duct system.

5 ● Where is direct humidification desirable?

Direct humidification is desirable when high humidity is required accompanied with cooling, ventilation, or air motion.

6 ● Where is indirect humidification desirable?

Indirect humidification is desirable when high humidity is required with simultaneous removal of heat by ventilation.

7 ● Why do cooling towers give best results when the humidity of the air is low?

The cooling of the water by dropping it through the air depends mostly upon the evaporation of the water. If the relative humidity of the air is low, the water vapor will be readily absorbed and carried away, while if the relative humidity is high, its capacity to pick up water vapor is less and the water is cooled less with the same exposure to the air.

8 ● What are some of the advantages and disadvantageous of a forced draft cooling tower compared with a natural draft wind tower?

Advantages: a. Does not depend on wind, *b.* Less space required, and *c.* Less drift loss and less make-up.

Disadvantages: a. Higher first cost, and *b.* Higher maintenance cost.

9 ● What wet-bulb temperature for outside air is usually selected in air conditioning design when cooling is to be accomplished?

One which is not exceeded more than 5 to 8 per cent of the time in the locality where the plant is situated.

10 ● Where should the suction connection be placed in a cooling pond?

As far below the surface as possible and as far away from the discharge as practicable.

11 ● What chemical is used to kill algae formation in spray ponds?

Potassium permanganate.

12 ● What is the usual amount of spray water delivered to a cooling pond per square foot of area?

From 0.1 gpm on small sizes to 0.8 gpm on large sizes.

13 ● About how much water is lost by evaporation in atmospheric cooling?

About 1 gal per 1000 gal for each degree of cooling range.

14 ● How is freezing obviated in cooling pond sprays?

The pressure and quantity of water is lowered so that the drops become larger in size and do not freeze so readily.

Chapter 26

AIR CLEANING DEVICES

Air Cleaner Requirements, Classifications, Viscous Type Filters, Unit Filters, Automatic Filters, Dry Air Filters, Air Washers, Methods of Installation, Stack Gases, Settling Chambers, Centrifugal Separators, Industrial Filters, Electrical Precipitators, Exhaust Systems, Air Scrubbers

THE removal of impurities from air brought into a building, or from air recirculated in a building for ventilating or air conditioning purposes is the function of any air cleaning or filtering device. These impurities include carbon (soot) from the incomplete combustion of fuels burned in furnaces and automobile engines, particles of earth, sand, ash, automobile tires, leather, animal excretion, stone, wood, rust and paper, threads of cotton, wool and silk, bits of animal and vegetable matter, bacteria and pollen. Microscopic examination shows that the character of the impurities varies with the locality, but as a rule carbon forms the greater part of them while the total is somewhat proportional to the state of industrial activity and the wind intensity. Additional information on sources of air pollution and the particle sizes of atmospheric impurities will be found in Chapter 4.

AIR CLEANER REQUIREMENTS

To fulfill the essential requirements of clean air, an air cleaner should:

1. Be efficient in the removal of harmful and objectionable impurities in the air, such as dust, dirt, pollens, bacteria.
2. Be efficient over a considerable range of air velocities.
3. Have a low frictional resistance to air flow; that is, the pressure drop across the filter should be as low as possible.
4. Have a large dust-holding capacity without excessive increase of resistance, or have ability to operate so as to keep the resistance constant automatically.
5. Be easy to clean and handle, cleans itself automatically, or else be inexpensive enough to replace when dirty.
6. Leave the air free from entrained moisture or charging liquids used in the cleaner.

The SOCIETY has developed a code[1] which explains how such devices are rated by (1) capacity in cubic feet of air handled per minute, (2) resistance in inches of water at rated capacity, (3) dust arrestance, the

[1]A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

percentage relationship expressing dust removal efficiency at rated capacity, (4) reconditioning power, the energy necessary to operate the mechanism of an automatic air cleaning device, and (5) dust-holding capacity, the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

CLASSIFICATION OF AIR CLEANERS

According to the Code, the following four classifications are given the devices:

Class A. Automatic Type: In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

Class B. Low Resistance Non-Automatic Type: Air cleaning devices for warm-air furnaces, unit ventilating machines and similar apparatus and installations in which a maximum of not more than 0.18 in. water gage is available to move air through the air cleaning device.

Class C. Medium Resistance Non-Automatic Type: Air cleaning devices for systems in which a maximum of not more than 0.5 in. water gage is available to move air through the air cleaning device.

Class D. High Resistance Non-Automatic Type: Air cleaning devices for the air intake of compressors, internal combustion engines, and the like, where a pressure of 1.0 in. or more water gage is available to move air through the air cleaning device.

Air cleaners may also be classified as follows:

1. According to principle of air cleaning.
   a. Viscous air filters.
      (1) Unit type.
      (2) Automatic type.
   b. Dry air filters.
   c. Air washers.
   d. Electrical precipitators.
2. According to application.
   a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type, as well as the non-automatic viscous unit or dry type are usually recommended and are installed in a central plenum chamber.
   b. For unit ventilators. Filters of viscous unit or dry type, installed at inlet of individual units.
   c. For window installations. Self-contained units consisting of fan and filter, usually dry or viscous type, adapted to be placed in the ordinary window.
   d. For warm-air furnaces. Unit type viscous or dry filters placed in small plenum chamber of warm-air house heating systems.
   e. For compressors and Diesel engines. Unit or automatic type viscous or dry filters, installed at air intake of compressors and Diesel engines.
   f. For compressed air lines. Unit type viscous or dry filters.
   g. For stack gases. Settling chambers, dynamic or electrical precipitators.
   h. For exhaust systems. All types.

Air cleaners may be classified further as follows:

1. For general air conditioning. With the growing congestion of large cities and an industrial growth throughout the entire country, the percentages of foreign material in the air, such as soot or carbon, which are unaffected by an air washer type of air cleaner, have increased. This has brought about the development of

the viscous and dry type air filters which are part of many ventilating and air conditioning systems.

2. For removal of dusts, smokes and fumes from stack gases. Prevention of atmospheric pollution from this source is of ever increasing importance, sometimes forced legally and frequently used in order to obtain increased efficiency.
3. For removal and collection of industrial dusts from the point of their production through exhaust systems.

VISCOUS TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement*. Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. While the arrangements of filtering media and the kind of materials used are almost unlimited, there are certain rather definite requirements for a practical commercial filter.

Investigations in this country and abroad demonstrate that the first impingement of dust laden air on a viscous coated surface removes about 60 per cent of the dust, the next impingement takes 60 per cent of what then remains—that is, 24 per cent—and the next impingement removes 9.6 per cent. To secure maximum efficiency, it is necessary to divide the air into innumerable fine streams, as the more intimately and freely the air is brought into contact with the viscous-coated media the better will be the cleaning.

The binding liquid used with viscous filters should have the following properties:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.
2. The viscosity should vary only slightly with normal changes of temperature.
3. It should be germicidal in its action to prevent the development of mold spores and bacteria on the filter media.
4. The liquid should have a high affinity for dust at low temperatures.
5. The liquid should have high capilarity, or ability to wet and retain the dust.
6. Evaporation should not exceed 1 per cent.
7. It should be fireproof.
8. It should be odorless.

Viscous Unit Filters

In the unit type viscous filter, the filtering media are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet. Where necessary reconditioning equipment should be installed near each group of unit filters, with hot water and sewer connections provided.

To secure greater dust holding capacity and a practically constant resistance and air volume, the filter media are usually placed in the direction of air flow, with progressively finer filter densities determined by the percentage of dust impinged. This arrangement provides relatively large spaces for the collection of dirt in the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while at the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impingement type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency of the unit is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity

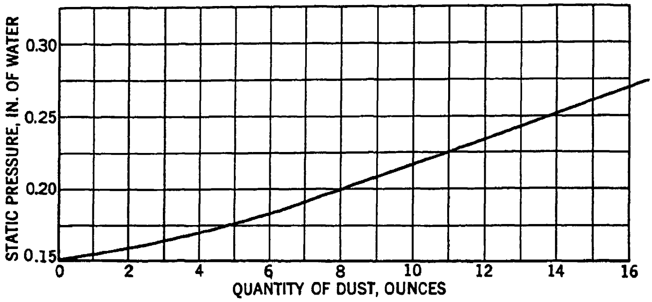

FIG. 1. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

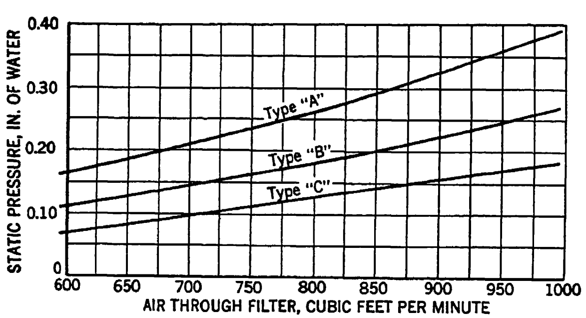

FIG. 2. RESISTANCE TO AIR-FLOW OF TYPICAL UNIT AIR FILTERS

of a built-up filter may be held at any desired figure. The frequency of cleaning any unit filter installation depends upon the dust concentration of air being cleaned, and on the amount of dirt which can be accumulated in the filter medium without causing excessive resistance. (Figs. 1, 2 and 3.)

It is difficult to satisfactorily compare the cleaning efficiencies of various filter types unless the efficiency ratings are determined under laboratory conditions in accordance with some definite test procedure such as that developed by the SOCIETY.[2] Efficiency tests made in the field with *atmospheric dust* are subject to so many variables that consistent comparisons are difficult. Of course there is no *standard atmospheric dust,* as atmospheric dust varies widely in composition and concentrations in different

[2]Loc. Cit. Note 1.

localities. Wide variations are also found due to different seasons of the year as well as the time of day and the direction of the wind. A chart showing the increase in resistance of a unit filter of the viscous impingement type, when tested with the standard test dust described in the code[3], is given in Fig. 1. The resistance to air flow of three typical clean viscous impingement type filters having different media densities is shown in Fig. 2. Type A is a dense pack used in bacteria control; Type B is a medium pack used for general ventilation work and Type C is a low resistance unit for use where low resistance is the important factor and maximum cleaning efficiencies are not essential. The operating characteristics which might be expected under various dust concentrations with air filters having different dust-holding capacities are illustrated in Fig. 3.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool, steel wool or the like are available.

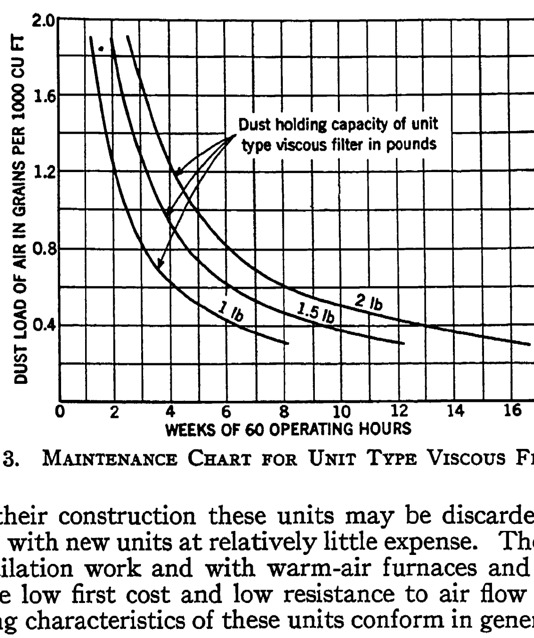

FIG. 3. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm-air furnaces and other installations where low first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

The principle of air cleaning used in the viscous automatic filters is the same as in the unit filters. The removal of the accumulated dust, however, is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter thus is deposited finally in the bottom of the viscous fluid reservoir from which it may be

[3]Loc. Cit. Note 1.

removed by different methods, depending on the design of the filter.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used by each type, as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain rotates slowly through this bath, thus performing the cleaning and recoating of the filter medium.

2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.

3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

The washing and renewing process in automatic filters usually is intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is ⅜-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or filtered through screens made of felt, cloth, or cellulose, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture of the filtering media used in most of the dry filters, the surface velocity, or velocity of the air entering the media, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, and the filter media are usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As in viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering media are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials as filter media usually depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter media should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

AIR WASHERS

Air washers have not been used extensively in the past in cleaning air for ventilating purposes because of their inability to remove fine dirt

particles. However, new types have been developed which appear to have possibilities for applications where the air to be cleaned is extremely dirty or where a higher degree of cleanliness is desired than can be obtained with a conventionally designed air washer. Information on air washers used in connection with humidifiers will be found in Chapter 25.

METHODS OF INSTALLATION

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 per cent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.
2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.
3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.
2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.
3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.
4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.
5. Electric lights should be installed in the chamber in front of and behind the air filter.
6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.
7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.
8. Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

STACK GASES

Solid particles discharged with stack gases, both domestic and industrial, contribute to the need for air cleaning in general ventilation. The common foreign matter includes the larger fly-ash and unburned carbon particles ranging up to 100 microns and larger, as well as the permanently suspended smokes. Usually it is economical to collect the coarser par-

ticles in separators, either gravitational or centrifugal, thus preventing clogging and overloading of the filters or precipitators used for the fines. Air cleaning devices for this purpose must meet the severe conditions of temperature and corrosion while handling large air volumes at low power and labor costs.

SEPARATORS

In addition to the air cleaning devices previously mentioned, the following are common types available for application to the removal of stack gases.

Gravitational Settling Chambers

The larger dust and gas particles will settle out from air if time and space are provided. Since the settling rate is constant, the required time of retention of the air in a gravitational settling chamber varies directly with the distance through which the particles must fall before reaching a retaining surface. Horizontal plates placed parallel with the air flow are effective and introduce negligible resistance. Air velocities should be selected so that the settled dust will not be redispersed, and for this reason baffles or constrictions producing increased velocity or turbulence should be avoided.

Relations between time of gas passage, distance of fall, and size of particles removed can be calculated from Fig. 1 in Chapter 4. With a forward air velocity of 50 fps passing between horizontal 14 ft shelves placed 3.3 in. apart vertically, particles of 100 microns, which settle at the rate of 59.2 fpm, will all have time to settle through the 3.3 in. vertical distance and reach the shelf while the air is passing along the shelf. Due to redispersion, actual operation would be much less favorable except at low air velocities.

Simple settling chambers consist of large spaces through which air velocities are decreased to one or two feet per second and in which dust particles fall into hoppers. In proper design, the inlets and outlets are placed and baffled so as to cause minimum turbulence, and the collected dust is protected from eddy currents.

Centrifugal Separators

The force causing settling can be increased many times that of gravitation by giving the air a whirling motion and introducing centrifugal force. The settling rate then becomes dependent upon the peripheral air velocity and the radius of curvature as well as upon the other factors.

In centrifugal and cyclone separators, air is introduced tangentially into a vertical cylinder and passes out from the center of the top. The gas velocity and curvature of the cylinder cause whirling which throws the particles to the surface. In the simple centrifugal type, the particles slide down the surface and are removed through a hopper in the cone bottom. In the cyclone type, they are thrown through slits in the periphery and collect in a second outer cylinder where the air is nearly static and there is little chance for redispersion.

Assumptions regarding streamline flow and turbulence make general calculations of centrifugal settling rate quite involved and rough. Their

range of usefulness is indicated in Fig. 1 of Chapter 4. They have wide application in connection with industrial operations such as grinding, screening, combustion, etc., but have little or no effect upon the finer particles.

Small diameters give smoother stream lines and larger centrifugal forces for the same power consumption, so that several small units in parallel are to be preferred to one larger one.

INDUSTRIAL FILTERS

In principle and practice the industrial dry filters are similar to those used for general ventilation, the latter being a development of the former. Bag filters up to 2.5 ft in diameter and 30 ft long, hung vertically, are fed through a header, allowing gas to pass out through the sides of the bag and retaining the dust particles on the inner surface. Depending on the nature of the cloth or mat filtering medium, retention of fines can be very high if gas velocity is low, about 0.5 to 3 cu ft per square foot per minute. The collected dust particles themselves aid in agglomerating and retaining others. Periodic shaking, with the fans off or reversed, at intervals of a few hours drops the excess dust into a lower header or hopper for removal.

Readily removed filters built in small sections in which filter media can be replaced are of distinct advantage where deterioration is rapid. Various styles of construction are available which combine quick interchangeability and large filtering area per square foot cross-sectional area. Use of several independent units in parallel is important for the reconditioning of each unit separately. Both continuous and intermittent shaking and sweeping devices remove excess dust and maintain a low resistance.

ELECTRICAL PRECIPITATORS

For removing fine dust or liquid particles which show no gravitational settling tendency, electrical precipitators are highly effective in air cleaning applications. In this system of air cleaning the particles are first ionized in a region where they acquire an electrostatic charge. The separation of the particles is then accomplished by passing the air between parallel plates where the dust particles are attracted to grounded collecting electrodes. The electric field holds them to this electrode unless a high critical redispersing gas velocity is exceeded. Particles are shaken down by either periodic mechanical or hand rapping.

The materials of construction for the apparatus may be selected to meet nearly any required conditions of temperature and corrosion. The discharge electrodes are usually of metal in the form of wires or edges placed equidistant between collecting electrodes either in the form of hollow pipes or plates. By properly choosing the type of electrodes the generation of oxides of nitrogen may be practically eliminated.

Voltage requirements depend primarily upon the electrode spacing and the gas conditions or particle nature. The maximum field intensity is limited by the arcking voltage for the particular conditions. The discharge electrode should be negative because with this charge higher voltages may be carried without arcking.

EXHAUST SYSTEMS

Quick removal of dust particles produced by such operations as grinding, screening, mixing, etc., is accomplished with exhaust systems. Their applications are numerous and varied, and not only prevent health hazards but eliminate product contamination. Mere discharge to the atmosphere outside of the building without collection is frequently of little effect, for incoming air redistributes the objectionable material. Information on the design of industrial exhaust systems will be found in Chapter 34. Any of the air cleaning devices previously described may be used with them depending upon the severity of the conditions and the nature and size of the material to be collected.

AIR SCRUBBERS

Air scrubbers are used extensively in exhaust systems since they provide removal of at least the coarser particles. The choice of scrubbing medium depends upon the character of the particles to be removed. The liquid medium should wet the particles, and the wetting is a surface tension phenomenon specific for each liquid-solid pair. Water effectively wets particles similar to silica, and oil wets particles similar to carbon. Combinations of oil and water, producing a froth, are effective for both and for materials of intermediate nature.

Intimate contact between the scrubbing liquid and the particles is essential, and many variations in constructions are available. Fine sprays, baffles, bubble caps, open and packed towers, and splash systems are used. Even the finest sprays are of low efficiency when used in an open chamber. Impact of the dust particles against a wetted surface is necessary for their retention, and this requires high gas velocities and well placed baffles or packing. Atomization of the liquid and air together is highly effective in removing the finest particles but makes for high power requirements.

Corrosion is frequently serious, particularly with high temperature gases containing soluble constituents. The collected material is removed as a thick sludge, and its wetted condition is a factor for consideration if it has possible recovery value.

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PROBLEMS IN PRACTICE

1 ● Assume a fan and duct system which handled 10,000 cfm through clean filters with a system resistance of 0.8 in. of water and that after the filters have become dirty the system resistance increases to 1.0 in. of water, and that the fan speed remains unchanged. Is there any way of predicting the volume of air delivered after the filter becomes dirty?

Yes. If the performance curves for the particular make of fan are available, the new volume may be determined from the resistance pressure curve. (Figs. 1, 2, 3 and 4, Chapter 27.)

2 ● What are the advantages of viscous filters?

The principal advantage of the viscous filter is its large dust holding capacity. The dust accumulation is distributed through the depth of the filtering medium rather than upon the surface as in the dry types, which makes it possible for viscous filters to handle heavy dust concentrations without excessive resistance. Since its efficiency and resistance are based on maximum air velocities of from 300 to 500 fpm through the filter, the viscous filter consumes the minimum amount of space for a given air volume.

3 ● What are the advantages of dry filters?

Dry filters are more efficient in the removal of fine dust particles from the air, and some types will eliminate even as much as 60 per cent of the smoke particles. Dry filters also are easily and conveniently maintained by vacuum cleaning, vibrating, or renewing the filtering medium.

4 ● If an air washer is used for cooling and humidity control in an air conditioning system, is a filter needed?

An air filter is desirable in conjunction with an air washer because of the large amount of soot in the air which, due to its greasy and amorphous nature, is not readily trapped in an air washer. Filters should be placed between the washer and the air intake so that all the dirt will be collected at one point to simplify maintenance and to protect all the equipment in the system.

5 ● Is an air filter needed with an extended surface type heat exchanger?

An air filter is essential with an extended surface heat exchanger in order to maintain its efficiency, for without this protection dust particles will adhere to the exposed surfaces, and gradually build up a deposit to the point where the efficiency will be impaired and the resistance increased by restricting the air passage.

6 ● What is the proper location of a filter in relation to the fan?

A filter will operate equally well whether placed on the suction or discharge side of the fan. It has become standard practice, however, to locate the filter on the fan inlet side because there it has: (1) simpler duct connections, (2) reduced static pressure losses, (3) more even air distribution over the entire filter area. Where an exceptionally high efficiency in dust removal must be maintained, it is often advisable to place the filter on the discharge side of the fan so there can be no infiltration of unclean air.

7 ● What instruments and apparatus are required for determining the pollen concentration in air by means of the settling method?

A microscope with a field of known area and a glass slide coated with a viscous material.

8 ● Describe the procedure for determining the pollen concentration in air by means of the settling method.

A glass slide coated with a viscous material is placed for a period of 24 hours in a horizontal position in the atmosphere to be tested. The slide is then removed and placed under the microscope, and pollen counts are made of approximately 25 fields over the area of the glass slide. Having determined the count over a definite area, as for example, 1 sq cm, and finding the settling rate of the average particles from the chart, Fig. 1 in Chapter 4, the concentration in parts per cubic yard can be calculated.

9 ● The resistance to air flow of a unit air filter is found to be 0.4 in. of water. The volume of air passing through the filter is 1000 cfm at a velocity of 200 fpm. What would be the filter area required in order to reduce the pressure drop across the filter from 0.4 in. of water to 0.16 in. of water?

Referring to Fig. 2: The resistance is substantially proportional to the square of the velocity, or

$$\frac{R_1}{R_2} = \frac{V_1^2}{V_2^2}$$

$$\frac{0.4}{0.16} = \frac{200^2}{V_2^2}$$

$$V_2 = 126.5 \text{ fpm}$$

$$Q = AV$$

$$1000 = 126.5\,A$$

$$A = \frac{1000}{126.5} = 7.91 \text{ sq ft}$$

The filter area would be increased from 5 sq ft to 7.91 sq ft.

10 ● A ventilating system complete with filters has a fan which, when operating at 400 rpm and delivering air at 1 in. of water total static pressure, requires an input of 3 horsepower. After the system operates for a time, the pressure drop across the filter caused by the clogging action of the collected dust and dirt increases from 0.1 in. of water to 0.4 in. of water. To maintain the original rate of air delivery with the increased static pressure, at what speed must the fan be run and what horsepower will be required?

Static pressure after clogging of filter $= 1 + (0.4 - 0.1) = 1.3$ in. of water.

The static pressure varies as the square of the fan speed. Therefore, if X is the fan speed after the static pressure increases:

$$\frac{1.3}{1} = \left(\frac{X}{400}\right)^2$$

$$X = 456 \text{ rpm.}$$

The horsepower varies as the cube of the fan speed. Therefore, if Y is the horsepower after the static pressure increases:

$$\frac{Y}{3} = \left(\frac{456}{400}\right)^3$$

$$Y = 4.44 \text{ horsepower.}$$

To maintain the original rate of air delivery with the increased static pressure, the fan speed must be increased from 400 to 456 rpm, and the horsepower from 3 to 4.44.

Chapter 27

FANS

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Volume Control, Fan Designations, Motive Power

IN heating and ventilating practice, fans are used to produce air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where an axial flow fan is intended for operation at comparatively high pressures the hub sometimes is enlarged in the form of a disc and the fan is known as a *disc fan*.

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan; a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances, especially where no long ducts or heavy friction must be overcome and where noise is not objectionable, whereas centrifugal fans are commonly employed for operation at the comparatively higher pressures and where extreme quietness is necessary.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These

laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.07492 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.06015 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.06015}{0.07492} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.06015}{0.07492} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and power vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.07492}{0.06015}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.07492}{0.06015}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.07492}{0.06015}} = 4.46 \text{ hp}$$

6. For a constant weight of air:
   (*a*) The speed, capacity, and pressure vary inversely as the density.
   (*b*) The horsepower varies inversely as the square of the density.

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.07492}{0.06015} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.07492}{0.06015} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.07492}{0.06015} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.07492}{0.06015}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^{1} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \tag{1}$$

When the static pressure is used in the computation it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static efficiency}^{1} = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{2}$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total or dynamic pressure. This efficiency is variously known as the total, dynamic, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total efficiency}^{1} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{3}$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different per-

[1]See Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, Edition of 1932.

centages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[2] as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers*. The results of tests are plotted in different ways: the

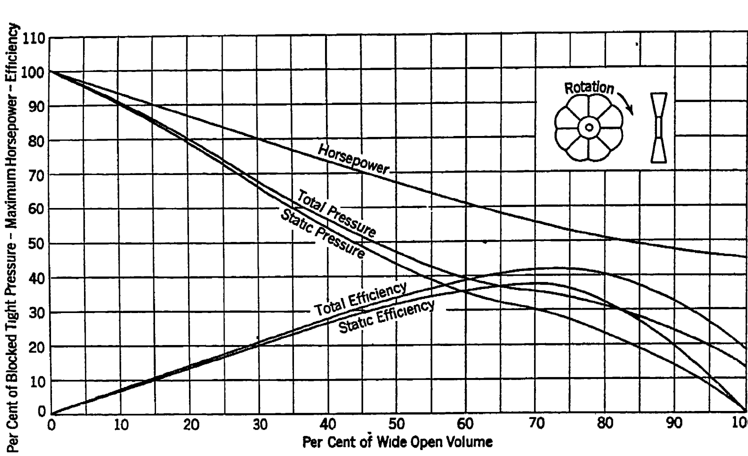

FIG. 1. OPERATING CHARACTERISTICS OF AN AXIAL FLOW FAN

abscissae may be the ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. Figs. 1 to 4 show characteristic curves for different types of fans using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and more particularly to show the manner in which variations occur with changes in fan capacity.

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[2]A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407. Amended June, 1931.

Axial flow fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in power as the resistance increases. The low efficiency when overcoming

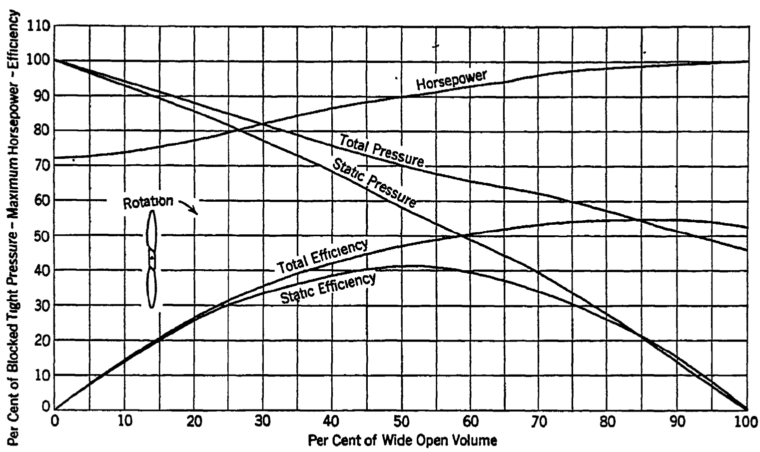


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the horsepower curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch, camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

Another advantage of this type of axial flow fan is its low resistance to air passage when standing still. There are some installations in which such a characteristic is desirable.

The straight blade (paddle-wheel) or partially backward curved blade type of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

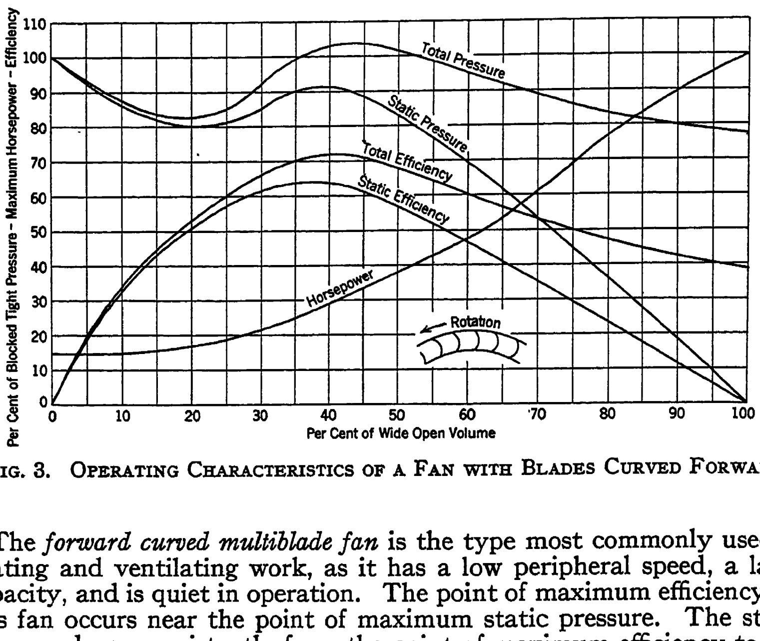


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity, and is quiet in operation. The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4.) This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type

makes it adaptable for direct connected electric motor drives. The high speed may necessitate somewhat heavier construction and more operating attention or service. The dimensional bulk for a given duty often is 150 per cent of that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type centrifugal fans a number of modified designs exist, differing in the angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade type with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show

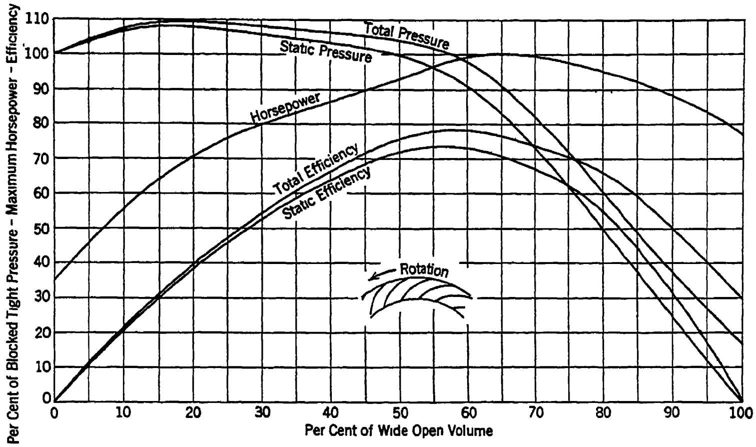


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

varying degrees of resemblance to the curves of Figs. 3 and 4, according to the degree of similarity to one or the other of the two designs of fan considered.

SYSTEM CHARACTERISTICS

A given fan performs as determined by the real characteristic of the system to which it is attached. When a different performance of a fan is desired, it is necessary to either change the speed of the fan (as *A* to *B* or *C* to *D* in Fig. 5), or to change the system (as by moving a damper from *A* to *C* in Fig. 5). If the speed of the fan is changed, the new point of operation is the intersection of the constant speed static pressure—cubic feet per minute curve for the new speed with the system characteristic. If the system is changed, the new point of operation is the intersection of the constant speed static pressure, cubic feet per minute curve with the new system characteristic.

Heating and ventilating systems follow the simple parabolic law quite closely but other types of systems follow some other more or less complex relation. The more complex systems can be separated into their component parts whose individual characteristics are known and the summation of the characteristics of the several parts of a system will give the composite characteristic of the system.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

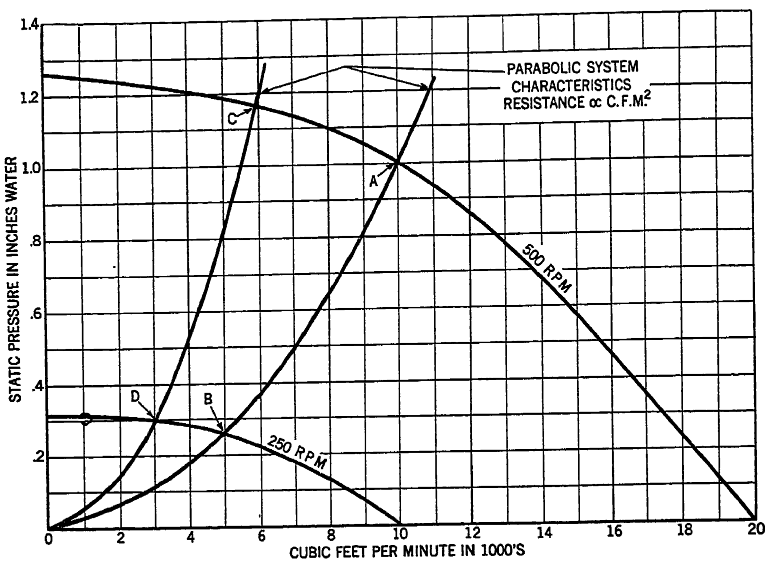


FIG. 5. ILLUSTRATION OF OPERATING POINTS OF A GIVEN FAN AT TWO SPEEDS ON THE SAME AND DIFFERENT SYSTEMS

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) speed, (3) noise, (4) size and weight, and (5) cost.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures:

1. Volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake power.
5. Tip or peripheral speed.
6. Static pressure.

The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Fans for Ventilating and Air Conditioning Systems

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Because noise in a ventilating system is irritating and a cause for complaint, fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls, and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed required for a specified capacity and pressure varies with the type of blade, and that a tip speed which may be excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR FORWARD CURVED MULTIBLADE VENTILATING FANS

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
1/4	1000–1100	1520–1700
3/8	1000–1100	1760–1900
1/2	1000–1200	1970–2150
5/8	1100–1300	2225–2450
3/4	1200–1400	2480–2700
7/8	1300–1600	2660–2910
1	1500–1800	2820–3120
1 1/4	1600–1900	3162–3450
1 1/2	1800–2100	3480–3810
1 3/4	1900–2200	3760–4205
2	2000–2400	4000–4500
2 1/4	2200–2600	4250–4740
2 1/2	2300–2600	4475–4970
3	2500–2800	4900–5365

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan is selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed in Tables 1 and 2 for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with the design of wheel and with the number of blades or vanes in the wheel.

Lower outlet velocities than those listed in Table 1 may be employed, but care must be exercised to avoid selecting a fan for operation below its useful range. The useful range of the fans of Table 2 extends over the full length of the performance curve.

In exhaust ventilating systems where the air column moves toward the fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building as when the air column is moving toward the rooms. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Amply large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers require definite increments of the total pressure which a supply fan in a ventilating system must overcome. These static pressures should be considered when selecting the fan characteristics, speed, and power.

TABLE 2. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS WITH BACKWARD TIPPED AND DOUBLE CURVED BLADES

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
1/4	800–1100	2600–3100
3/8	800–1150	3000–3500
1/2	900–1300	3400–4000
5/8	1000–1500	3800–4500
3/4	1100–1650	4200–5000
7/8	1200–1750	4500–5300
1	1200–1900	4800–5750
1 1/4	1300–2100	5300–6350
1 1/2	1400–2300	5750–6950
1 3/4	1500–2500	6200–7550
2	1600–2700	6650–8050
2 1/4	1700–2800	7050–8550
2 1/2	1800–2950	7450–9000
3	2000–3200	8200–9850

Fans picked within the limits of Table 1 will operate close to the point of maximum efficiency. No attempt has been made to select these limits for quiet operation, since this is a relative term and varies with the type and location of the installation.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where noise prevention is a factor the fan and its driver should have floating foundations.

Fans for Drying

Both axial flow and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing axial flow fans are widely used in the drying

field. In drying, these fans may be used with unit heaters where not too much duct work is required and where air is to be delivered against pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work.

Centrifugal fans of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes of air against all pressures likely to be encountered.

Belt driven fans usually are to be preferred to direct-connected fans since efficient motor speeds do not usually coincide with efficient fan speeds. Replacement of a standard motor is quick and easy if it is belted.

Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers. A throttled outlet reduces the volume and reduces the power.

Due to the low speeds of forward curved multiblade or paddle-wheel type fans, these can be direct-connected to reciprocating steam engines, and the exhaust steam from the engines may be used in the heating apparatus. In selecting engine driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a dryer varies, and where several fans must operate in parallel, fans are to be preferred which have a continuously rising pressure characteristic, such as is given by backward-curved or double-curved blades. This type of fan is well adapted for direct-connected motors of the higher speeds. (See Chapter 35 on Drying Systems.)

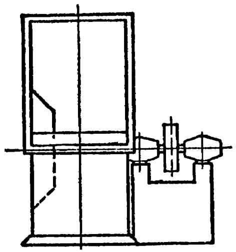
Fans for Dust Collecting and Conveying

The application of fans for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 34. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained resistance or total static pressure at which the fan is to operate. The selection of a proper size fan is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

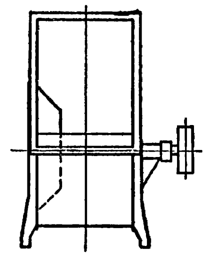
Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN VOLUME CONTROL

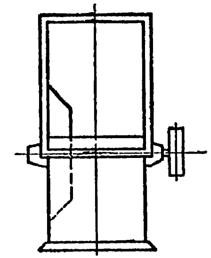
Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity or by interposing resistance, as by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an



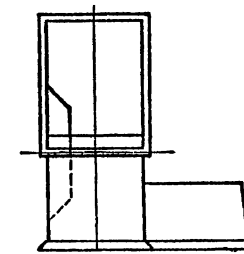
Arr. 1.
For belt drive.
Wheel overhung. Bearings on pedestal.



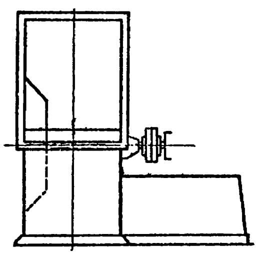
Arr. 2.
For belt drive.
Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.



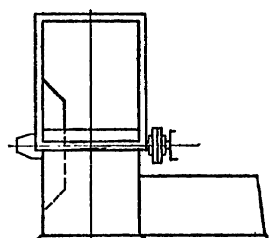
Arr. 3.
For belt drive.
Pulley overhung. Bearings supported on fan housing.



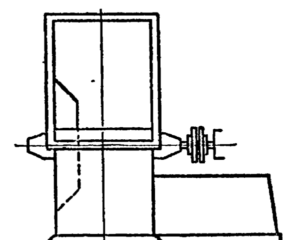
Arr. 4.
For direct drive.
Wheel overhung. No bearings on fan. Wheel mounted on motor or engine shaft. Pedestal for motor or engine.



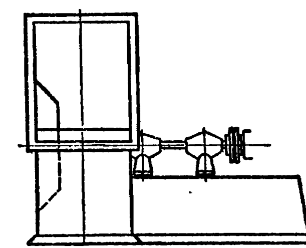
Arr. 5.
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.
Similar to Arr. 6, but with two bearings on fan, and flexible instead of rigid coupling.



Arr. 8.
For direct drive.
Similar to Arr. 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling.

FIG. 6. ARRANGEMENT OF FAN DRIVES

increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it may be exceedingly desirable to reduce the speed at times, while on the other hand, any fan which has its normal speed reduced as much as 50 per cent without *change in resistance* will move only 50 per cent of the air.

FAN DESIGNATIONS

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[3]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word "hand," but specify "clockwise" or "counter-clockwise."

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 6 are suggested.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan or, having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved* blade types are used, motors with ratings very close to that of the fan horsepower demand can be employed, provided the fan has a limiting horsepower characteristic.

[3]Recommendations adopted by the *National Association of Fan Manufacturers*.

Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.*

Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines, and turbines, but as previously stated the drive commonly used is the electric motor.

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PROBLEMS IN PRACTICE

1 ● What information must be supplied to the manufacturer when ordering a centrifugal fan?

a. Size of fan (catalog number).

b. Type of fan.

c. Width of fan (single or double).

d. Number of inlets (single or double).

e. Fan performance and kind of application.

f. Direction of rotation (clockwise or counter-clockwise).

g. Direction of discharge (top horizontal, down blast, etc.).

h. Drive arrangement (see Fig. 6).

i. Style of housing (full, three-quarters, etc.).

2 ● In selecting fans for quiet operation in public buildings:

a. Should the outlet velocity of the fan be limited?

b. Should the tip speed of the fan be limited?

a. Because all commercial fans operating at pressures suitable for this class of work would be considered noisy if the fan were to discharge directly into the room, and because the duct system on the fan discharge is depended upon to absorb a reasonable amount of fan noise, it is desirable to have a moderate run of duct work with some bends and elbows included as sound deadeners. Where this duct is of necessity very short, the outlet velocity must be kept down to the lower limits recommended in this chapter or else an efficient sound absorber must be used. The experience of the engineer must be his guide in determining the allowable outlet velocity in each individual case.

b. Tip speed should not ordinarily be limited, because different types of fan blades have entirely different allowable tip speeds for quiet operation. A fan having a backward blade at the tip can run at much higher tip speed than can a forward curved or a straight blade fan, with the same degree of quietness.

3 ● Is a direct connected or a belted fan preferable in public building work?

Where space is at a premium, direct connection is best. Next in space economy is the short V-belt drive. The flat belt drive fan requires the greatest floor space. In this class of work, pressures are usually so low that even with the high speed fans the motor cost is greater for direct connected units than for belt drive fans.

4 ● a. What type fans are used in industrial work?

b. What outlet velocity is suitable?

a. All of the centrifugal types are suitable; the disc and propeller types are suitable for low pressure work, or they are often used as exhausters.

b. The outlet velocities on fans for industrial work can be much higher than can those in public building work, where quietness is essential. Fans should be selected with outlet velocities as recommended in this chapter, using the upper limit of velocities.

5 ● Are direct connected or belted fans preferred in industrial work?

In industrial applications, fans are often advantageously direct connected to motors. The pressures are usually high enough to use standard motor speeds. The high speed types of fans have limiting horsepower characteristics so that little margin in power must be provided in the driving motor. Belted fans may be used, but where high power is required a special arrangement is often necessary for shaft and bearings on account of the weight of the sheave and the belt pull.

6 ● A forward curved multiblade fan which requires 5.4 bhp is delivering 22,800 cfm at 70 F against a resistance pressure of 1 in. of water at an outlet velocity of 1440 fpm:

a. What is the static efficiency?

b. What is the total efficiency?

a. 66.3 per cent (see Equation 2).

b. 74.5 per cent (see Equation 3).

7 ● If the above fan has a 54-in. diameter wheel and operates at 193 rpm, will it be suitable for a ventilating installation where a minimum of noise is desirable?

Yes. The tip speed will be 2720 fpm and this, together with the 1440 fpm outlet velocity, falls within the limits given in Table 1 for 1-in. resistance pressure.

8 ● What objectionable feature is inherent in the ordinary propeller fan when it is operating at high resistance pressures?

It must operate at a high speed with consequent noise.

9 ● At what point should a fan be selected for operation, and why?

At its point of maximum efficiency because the cost of operation and the noise produced will be least.

10 ● In Fig. 3, a static pressure of 85 per cent of blocked tight pressure corresponds to three different volumes, namely 11 per cent, 30 per cent and 48 per cent of wide open volume. What will determine which volume the fan delivers?

The fan can operate only at the intersection of its pressure-volume curve and the system characteristic. The type of system, together with the specification of the volume at a certain static pressure, completely defines the system characteristic.

As illustrated in Fig. 5, a given system characteristic will intersect the fan curve in only one point.

If the 85 per cent value for static pressure is specified for the 48 per cent value of volume, it is at once obvious that the same system will not have the same resistance at any other volume.

Chapter 28

AIR DISTRIBUTION

Definitions, Grille Locations, Standards for Satisfactory Conditions, Factors Affecting Distribution for Cooling and Heating, Air Outlet Noises, Selection of Supply Outlets, Balancing System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. Supplying the proper amount of air is one problem; properly distributing it from the point where it leaves the fan is another. The distribution problem may be further divided into: (*a*) distribution *to* the various spaces served by the system, (*b*) distribution *in* these spaces. This discussion is primarily limited to division (*b*), reference being made to the duct system only insofar as it affects the performance of the air distribution outlets.

Definitions

In this discussion, the term *air outlets* or *outlets* will be used to designate a cover for an opening, whether it is a grille or a register.

A *register* is defined as an outlet with a damper, and a *grille* is defined as an outlet without a damper.

The perpendicular distance over which the air will satisfactorily carry measured between the face of the outlet and the opposite wall is the *throw*. In the case of directional flow outlets, this may be less than the actual carry of the air.

The *core area* of the outlet is the area of that portion of the grille inside the frame through which the air can flow.

The ratio of width to height of the core area is termed the *aspect ratio*.

GRILLE LOCATIONS

The location of supply and exhaust outlets is extremely important if a satisfactory installation is to be secured. Very frequently, however, the room or building is planned and constructed with practically no consideration of this problem. The engineer of today is more likely than not to have as his problem a building that was constructed long before any consideration whatever was given to air conditioning it. Consequently, the room shapes, the location of columns and beams, and other details of architecture frequently make it difficult to properly locate the outlets. In general, for a cooling installation, the grilles should be located high enough from the floor to prevent the discharge of air directly upon the occupants of the room, and far enough down from the ceiling to minimize

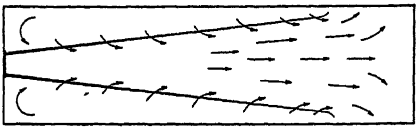

FIG. 1. PLAN VIEW LONG THROW SUPPLY OUTLET

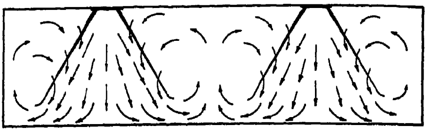

FIG. 2. PLAN VIEW SHORT THROW SUPPLY OUTLETS

the possibility of streaking, and to permit induction of air from all sides of the stream. If the stream actually strikes the ceiling, but at a small angle, the throw will be increased somewhat if the ceiling is smooth. If the angle at which the stream hits the ceiling is 20 deg or more, or if the flow along the ceiling is obstructed by panel mouldings or beams, air velocity may be rapidly lost and a decreased throw result. The air stream also should be so directed that it will not strike nearby columns or beams in such a way as to cause misdirection of the air stream or drafts. Where the room is of irregular shape, as an ell, or where it has an alcove in one side, consideration should be given to obtaining satisfactory circulation in these corners. Frequently this cannot be done except by the use of multiple supply outlets. In using multiple outlets, care must be taken that the several air streams do not interfere with each other, until their velocities have been reduced to values which will not cause high turbulence and a drafty condition. Beams and offsets in the ceiling will cause little difficulty when substantially parallel to the direction of flow, unless they are of considerable depth, but when positioned across the air stream, may cause drafts and failure to secure satisfactory circulation in that portion of the room farthest from the outlet. In the case of a heating installation, down-drafts produced by such obstructions may not be serious, because the air will rapidly lose its downward motion, but the possibility of failure to obtain satisfactory circulation still exists.

The location of supply outlets should, if possible, be such as to take advantage of the maximum velocity permissible from a noise standpoint. For instance, the spaces illustrated in Figs. 1 and 2 may be satisfactorily served by either arrangement. However, by taking advantage of the long

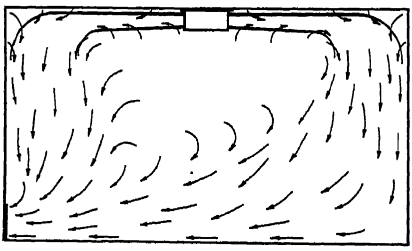

FIG. 3. ELEVATION VIEW CEILING SUPPLY OUTLET WITH RETURN WALL OUTLET

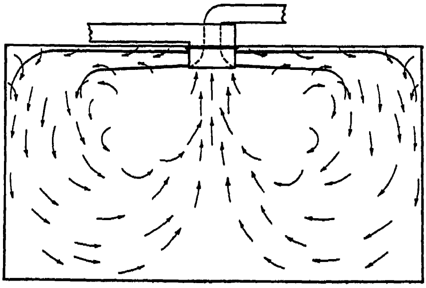

FIG. 4. ELEVATION VIEW CEILING SUPPLY AND RETURN OUTLET

throw, to which the arrangement in Fig. 1 lends itself, fewer outlets are required and additional savings are effected in the sheet metal work.

In solving the problem of properly conditioning a room of irregular shape, where multiple wall supply grilles are objectionable, a ceiling outlet of the type illustrated in Figs. 3 and 4 may very often be the best solution.

In choosing the most desirable location for the return air grille, consideration should be given to its effect on circulation of the air through the room. It is generally true that the return air grille should be placed on the same wall as the supply and near the floor level. This results in a

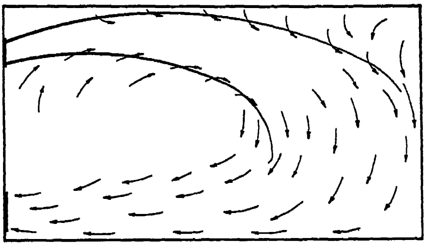

FIG. 5. ELEVATION VIEW CORRECTLY LOCATED RETURN OUTLET

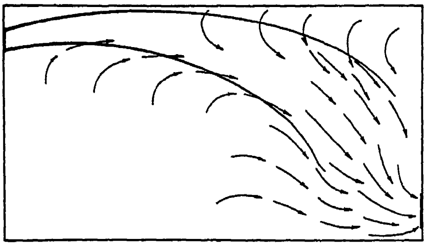

FIG. 6. ELEVATION VIEW OF IMPROPERLY LOCATED RETURN OUTLET

U-shaped air path (Fig. 5) which covers the room thoroughly. The arrangement shown in Fig. 6 should be avoided, because it tends to create a stagnant section below the supply grille. What would otherwise be an unsatisfactory dead spot in a room may in some instances be taken care of by location of the return air grille near that area (Fig. 7).

STANDARDS FOR SATISFACTORY CONDITIONS

The most satisfactory air condition cannot be definitely stated for any particular individual without conducting a series of tests with that individual as subject; some persons are less sensitive than others to variations in temperature, humidity, air velocity and noise. The best

that can be done is to attempt to set limiting conditions leaning toward the values of these variables which produce a condition of comfort for the greatest number of individuals. On a cooling installation, the allowable deviation from average room temperature, that is, the temperature of puffs of air which may strike a person momentarily, is a function of the room temperature as well as the velocity of the air. For instance, in a room controlled at 72 F, a puff of air at 70 F might be uncomfortable to an individual, even at relatively low velocities, whereas if the average room temperature were 80 F, air at 78 F, even at moderate velocities, might be very satisfactory. However, air at 78 F in an average room temperature of 83 F would be cold. In general, other conditions being equal, for the range of temperatures normally encountered in living quarters on cooling installations, the permissible deviation from average room temperature varies from approximately 1 F at the low end of the range to about 3 F at the high end of the range. In this matter, it is important to consider the particular problem in the light of the type of occupancy. For instance, greater deviations from room temperature and higher velocities may be permitted in a garage or a hotel hallway than would be permissible in an office or living room. The velocity which may be considered the permissible maximum differs with the temperature deviation for a given installation, but an absolute maximum under any conditions might be considered that which would produce a mechanical disturbance, such as the movement of a person's hair or disturbance of papers on a desk. Humidity is an important consideration in the determination of one's feeling of comfort; however, if the room generally is assumed to be at a satisfactory value of relative humidity, the designer is justified in neglecting this factor when considering permissible fluctuations in temperature and velocity in the occupancy zone. This is true because the maximum allowable temperature fluctuation results in an unnoticeable humidity change.

The standards that might be set up for maximum allowable room temperature deviation and air velocity would not be the same for both heating and cooling installations. In the former case, any appreciable temperature deviation is likely to be above rather than below the average room temperature, whereas the reverse is most likely to be true on a cooling installation. Further, because air movement has a cooling effect in itself, the feeling of warmth due to temperatures above room temperature is counteracted to a certain extent so that an individual may be subjected to higher velocities of warm air without the feeling of discomfort occasioned by the same velocities of cool air. In every case, it should be the purpose of the designing engineer to keep the conditions within the zone of occupancy as nearly uniform as possible, securing minimum temperature deviations and low velocities. The air velocity at all points in the room should be at least 25 fpm for good results.

It is impractical to measure momentary temperature differences with any degree of accuracy in the field, but in checking a given installation it will generally be found satisfactory to measure velocity only, since on cooling installations high velocities normally occur with low temperatures, and on heating installations high velocities occur with high temperatures. That is, in the former case, the chilled supply air loses its velocity and undergoes an increase in temperature as it settles into the occupancy

zone, whereas in the latter case the heated supply air loses its velocity and undergoes a decrease in temperature during this process. Therefore, if the average velocities within the occupancy zone are not excessive, one is fairly safe in assuming that the temperature difference is also within permissible limits.

The subject of sound control is covered in Chapter 30 and it is recommended for detailed review before consideration of the problem of air outlet noise. An understanding of the relation between sound intensity and loudness level in decibels, as well as the effect of the presence of sound absorbent materials in the room, is particularly necessary. A more detailed discussion of the nature of this problem appears later; whereas the following comments refer to what constitutes a satisfactory noise condition.

Obviously, the nature of the conditioned space is important when considering the allowable outlet noise. In factories, press rooms, and similar

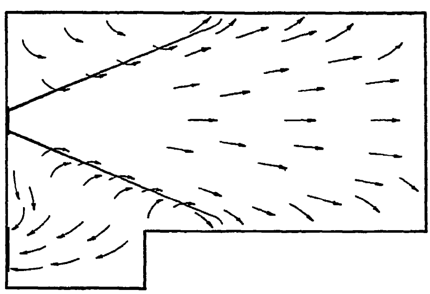

FIG. 7. PLAN VIEW CORRECTLY LOCATED RETURN OUTLET ELIMINATING STAGNANT SPACE

spaces where the noise level is 65 db or higher, no complaints of grille noise are likely to be made. On the other hand, some homes, offices, hospitals, and, most of all, radio broadcasting and movie sound studios present a real problem which must be intelligently attacked if a satisfactory installation is to be made. In this chapter the noise of the air outlets (and returns) only is considered, it being assumed that the noise or sound level of the room without the outlet noise includes that which may be contributed by fans, motors, duct work, and other items of conditioning equipment. The control of noise from these sources is another problem (see Chapter 30). Where sound control is important, the actual room sound level without conditioning equipment should be known. If feasible, the contribution of the conditioning equipment, less outlets, should be estimated to secure the working sound level. If this correction is not made, the use of the first value errs in the direction of safety.

It is evident that the point within the room which should concern the designer in this problem is that at which the outlet noise is greatest. A tentative standard *listening point* relative to the outlet is suggested later in this discussion, and it is assumed that the outlet noise data are taken

with reference to this point. If it is desired that the outlet noise result in an inaudible addition to the existing noise level, it is safe to assume the total outlet noise to be 5 db below room level. This results in an increase in total noise of slightly over 1 db, which is unnoticeable. If an increase of 3 db is permissible, the outlet noise level may be equal to the room noise level alone. All outlets in the room must be considered, as will appear later, and the returns may be ignored only if they are so sized that the velocity of air through them is much less than through the outlet.

DISTRIBUTION FACTORS IN ROOM COOLING

In attempting to design a satisfactory air distributing system, it is first necessary to properly locate the grilles in accordance with the recommendations already stated. Assuming that the best locations have been selected, it then becomes necessary to choose the proper grille for that location. The considerations involved are the amount of air to be handled, the velocity permissible from the standpoint of noise, and the distance the air should carry. The distance it will carry, assuming no obstructions, is affected by a number of factors which are listed below:

1. The temperature difference between incoming and room air.
2. Height of grille above floor.
3. Face velocity.
4. Core area.
5. Core aspect ratio.
6. Design of grille.

The manner in which the above factors affect throw may be generally stated. All other things being constant, a lower temperature of incoming air will result in shorter throw; a greater height above the floor will affect a longer throw; a higher velocity will produce a longer throw; greater area will give longer throw; larger aspect ratio will decrease throw. The variation in throw with type of outlet will, of course, depend upon the design characteristics of the outlet.

In consideration of what constitutes the possible throw of an outlet under a given set of conditions, it is important to remember that the throw may be unsatisfactory for any one of several reasons:

1. It may be so long that it will strike the far side of the room and come down the wall with velocities higher than are permissible,
2. It may be so short that it will fail to carry the full length of the room, and short-circuit to the return air outlet, or
3. It may spill into the center of the room.

In the first case, the system fails for lack of uniform distribution and the presence of cold areas. In the second case, the standards as to velocity and temperature difference in the zone of occupancy may be satisfactorily met, but air distribution and circulation throughout the entire room is not accomplished, with the result that the end of the room away from the outlet would not be satisfactorily conditioned. In the third case, the shortcomings of both case one and case two are present. It is evident, therefore, that for a given outlet discharging air at a given velocity, there is a maximum and a minimum length of room which can be satisfactorily

handled. In the latter, the velocity of the air down the far wall is just within the maximum permissible, while in the former, satisfactory circulation is barely accomplished.

In general, the higher the outlet is above the floor, the greater may be the difference between room air and incoming air temperatures.

Assuming that proper supply outlets for a given installation have been selected, unsatisfactory performance may still result due to the construction of the duct work immediately back of the outlets. Performance data on the grilles and registers of various manufacturers should be based upon results obtained with the air approaching the grille perpendicularly and at uniform velocity over the entire duct cross-section. Where this condition does not exist in practice, performance predictions based on published data cannot be expected to be realized. Every precaution should be taken to secure as nearly ideal conditions in the approaching air stream as are possible.

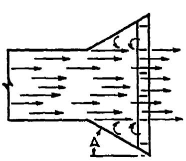

FIG. 8. EFFECTS OF EXPANDING DUCT

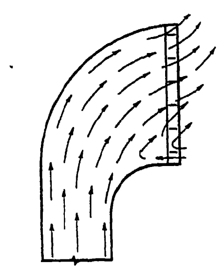

FIG. 9. UNEQUAL FACE VELOCITIES

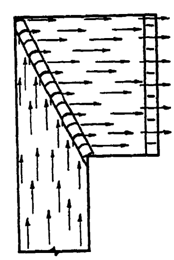

FIG. 10. EFFECT OF TURNING MEMBER

In addition to disturbances due to the construction of the duct work itself are those which may be created by dampers immediately behind the grille. Where either multiple louvre or single blade dampers are used, considerable deflection of the air stream may result, if it is throttled appreciably by these means. This is particularly true when the fins of the register core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

Any attempt to secure a low face velocity and high duct velocity by the construction of any expanding chamber immediately behind the grille is very likely to be unsuccessful. In order to expand from a small duct to a larger one, and have the air stream fill the duct at the end of the diverging section without turbulence, angle *A* in Fig. 8 should be about 3 deg for four-sided expansion and about 5 deg for two-sided expansion. From this it is apparent that an attempt to secure equivalent results with a short connection would be futile. What actually happens when this is attempted is illustrated by the arrows in Fig. 8. When localized high velocities through the outlet exist from this cause or any other, the noise produced will naturally exceed that which the outlet area and average face velocity would lead one to expect. This fact should be remembered

in considering the use of register dampers, particularly in those cases where there must be considerable throttling with the damper to balance a poorly designed system. Where reduction of noise is important, it is recommended that balancing dampers be placed in the duct *ahead* of the acoustic duct lining.

Similar unequal face velocities, aggravated by a deflection of the air stream, are obtained with the arrangement shown in Fig. 9. The latter may be corrected by inserting a turning member in the elbow back of the outlet face as shown in Fig. 10. The importance of straightening the air stream and affecting uniform distribution over the entire face of the outlet cannot be over-emphasized.

DISTRIBUTION FACTORS IN ROOM HEATING

The problem in the case of a heating installation is substantially the same as in cooling, with a few exceptions. Because the temperature of the incoming air is above that of the room, there is no tendency for it to drop and consequently the throw is not particularly affected by temperature difference in a low ceiling room. In general, the air should be deflected downward where the grille is above the occupancy zone, and this is particularly desirable where the ceiling is high. For the same reason, that is, to keep the heat in the occupancy zone and to avoid excessive temperature at the ceiling, it is desirable to have the grille comparatively low on the wall, and just slightly above the occupancy zone. If the grille is lower than this, it may create an unsatisfactory condition of very warm air at quite high velocities where it can possibly strike the occupants of the room. Where the velocities are very low, the grilles may even be satisfactorily located below the 6 ft level, although the immediate vicinity of the supply outlets will probably be useless for occupancy because of high temperature. Essentially, the problem is to keep the incoming air up for cooling, and down for heating, until it is thoroughly mixed with the room air. Grilles and registers which are adjustable for deflection upward and downward, either by moving the fins or inverting the grille, are in general use.

AIR OUTLET NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the outlet) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db. However, it is not satisfactory to consider the grille noise on this basis (wherein the sound power received directly from the source is small compared with that received by reflection) since in practice the occupants of the room may be quite close to the grille. The nearer the listener is to

the sound source, the greater the proportion of the sound intensity which is due to direct transmission.

In the absence of generally accepted standards at this time it is suggested that the loudness level 5 ft from the lower edge of the outlet, measured downward at 45 deg in a plane perpendicular to the outlet at its center, represents about the maximum within the zone of occupancy. The cases where persons are nearer to the outlet than this are rare and are ignored in the consideration of this problem. Although the effect of sound absorbent material on the intensity at the 5 ft station is not nearly so great as at more remote points in the room, it should not be ignored without consideration of the error involved. An average living room may contain 100 sabins (absorption units). If this be decreased to 50 sabins, the *diffuse* or reflected sound level would be increased 3 db. However, at the 5 ft station the increase would be less than 2 db. If the absorption of the room be increased to 200 sabins, one might expect a reduction in diffuse noise of 3 db; but at the 5 ft station the redu tion would be less than 1½ db. Furthermore, even though the absorption be increased without limit (as in free space) the reduction would still be less than 2 db because of proximity to the source.

In comparing sound ratings of various grilles, the following must be known if the information is to be intelligently applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level; if not, the room level (without grille noise) must be known.
6. Methods used for recording data.

Data mentioned in this chapter are assumed to have been referred to the following:

1. Threshold intensity = 10^{-16} watts per square centimeter[1].
2. Microphone location 5 ft from lower edge of outlet on a line downward at 45 deg and in a plane bisecting the outlet perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabins absorption.
5. Plotted data are loudness levels of *outlets only*, correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighting network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure to total sound level of outlets of more or less than one square foot area. This can be done by use of the following formula:

$$\text{Decibel Addition} = 10 \log_{10} A \qquad (1)$$

[1]American Tentative Standards for Noise Measurement, *American Standards Association.*

where: A = core area, square feet.

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. Since total loudness and air flow are both functions of velocity and area, the solution of the problem by use of the previous analysis implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves as illustrated in Fig. 11. With this chart it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular make of grille,

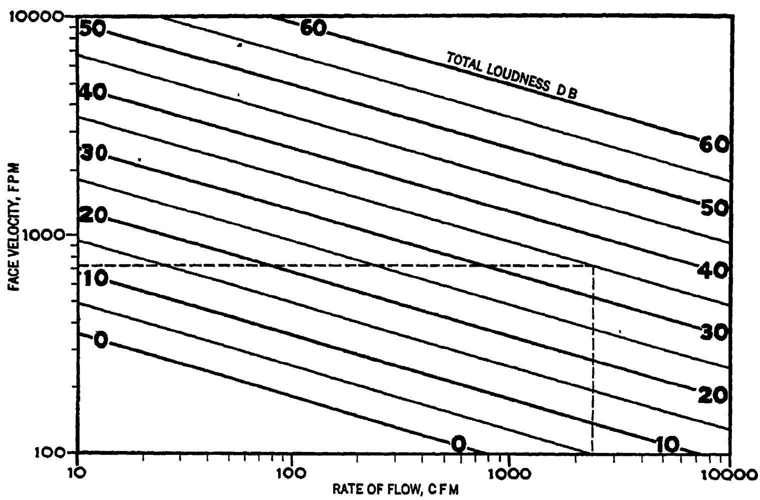

FIG. 11. AIR FLOW AND LOUDNESS CHART

register or air outlet. It is assumed that Fig. 11 is based on a room having 100 sabins of sound absorption. In such a room the sound level due to other sources may be 35 db. As previously stated an outlet having a noise level of 30 db would be substantially inaudible in such a room.

If 1000 cfm are required with a total noise due to outlet of 30 db, a velocity (Fig. 11) of about 675 fpm may be used. From this velocity and the rate of flow, the core area can be computed. This determination was on the basis of a room absorption of 100 sabins. If the absorption is greater, the 675 fpm velocity is safe, since the loudness level will go down. However, correction can be made if desired by the use of the chart of Fig. 12. Thus, if the absorption is 200 sabins, a correction of +1.3 db may be made and the permissible velocity becomes that corresponding to a total loudness level of 31.3 decibels or approximately 750 fpm. If the room is highly reflecting and has an absorption of less than 100, correction is much more important. For instance, for 35 sabins a correction of −3 db

must be made and the maximum velocity corresponding to 27 db total loudness chosen; that is, approximately 550 fpm.

Where more than one outlet must be considered, the problem is more complicated. If a similar outlet is added in a far corner of a highly absorbent room, the change in noise level at the 5 ft station at the first

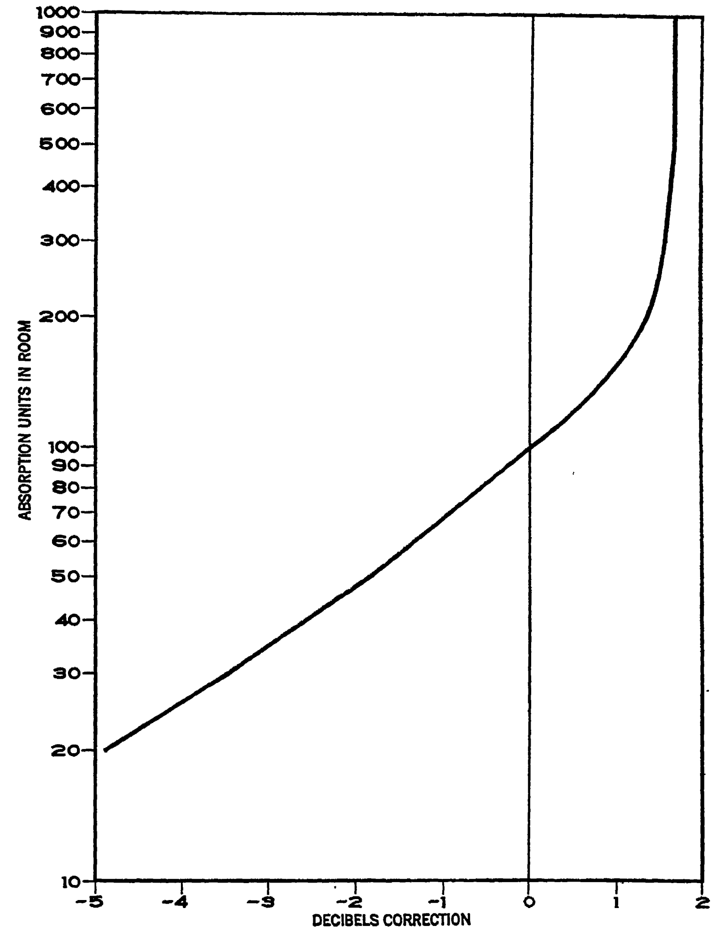

FIG. 12. ROOM ABSORPTION CORRECTION CHART

outlet is small; however, if the room is small, or highly reverberant (both, the intensity at the 5 ft station may be almost doubled and tl noise level increased nearly 3 db thereby. The simplest method of han(ling this problem, and one which errs in the direction of safety, is to tre: the room as though all the air were being supplied by one outlet. Thu if two outlets, each supplying 1000 cfm are used, the value 2000 cf

should be used with Fig. 11. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to make this penalty serious or to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply outlets. That is, if 1000 cfm is supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 11.

SELECTION OF SUPPLY OUTLETS

After the heating and cooling load calculations have been made (Chapters 7 and 8), and, or a suitable supply air temperature selected, the

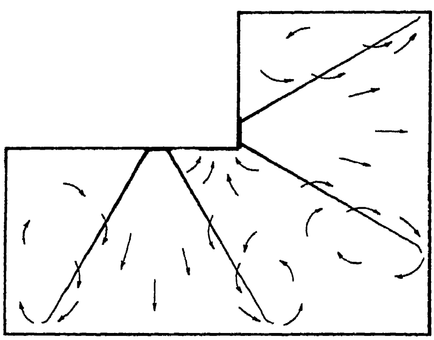

FIG. 13. PLAN VIEW TYPICAL GENERAL OFFICE

volume of air required for each space can be determined. The next step is to determine the velocity at which the air may be introduced into the space quietly and without creating objectionable drafts.

Present-day grille design coupled with the introduction of effective acoustical treatment for minimizing fan and duct noises have made grille face velocities in excess of 1500 fpm feasible, and 600 to 1200 fpm is now used in practice. This range of velocities is approximately three times higher than common practice values of a few years ago.

Since high velocities make for smaller ducts and outlets, and therefore savings in space as well as greater flexibility in locating the duct work to the best advantage, selection of design velocities is a very important step.

The selection of proper velocity requires that the designer have before him reliable data applicable to the particular make of outlet he proposes to use. Even under these circumstances, the problem is one of *cut and try* because permissible velocity may be determined by either noise or throw.

A method for selecting supply outlets is outlined below in the form of a sample cooling problem, using numerical values which have no reference to any particular make of outlet.

1. The load calculations have been made; a suitable temperature differential has been selected (it is to be understood that the data referred to from this point on are based on this temperature differential), and the volume of air required determined. Assume that Fig. 13 represents a small general office having a noise level of 40 db and that 2500 cfm must be supplied for proper conditioning.

2. Select a tentative location for the outlet or outlets, having in mind the type of grille most likely to effect proper distribution. In this particular case, two outlets having a wide spread appears to be a logical choice.

3. Data from which to determine velocity which corresponds to 2500 cfm and a noise rating at least 5 db below the noise level of the office may be presented in a number of forms, one of which is shown in Fig. 11. (Fig. 11 represents assumed values only. In practice similar data should be obtained from the manufacturer whose outlets are being considered. Several similar charts or tables may be necessary to cover any one manufacturer's complete line.) From Fig. 11 it will be noted that for 2500 cfm the type of grille selected may be used at velocities up to 700 fpm without exceeding 35 db; that is, 5 db below the noise level of office.

4. Having determined the velocity, the core area becomes fixed at 3.57 sq ft or 257 sq in. per outlet. In this problem, the two grilles in question are so close together that consideration of their combined area in determining the permissible velocity from the standpoint of noise introduces little error.

5. The type grille selected has thus far been found satisfactory from a noise standpoint, provided the face velocity does not exceed 700 fpm. The next consideration is throw, which may be assumed to be 16 ft, and by reference to a manufacturer's catalogue the proper correlative test data may be checked with the throw assumed. It is of course evident that one or more types of grilles may satisfy the requirements, and that in any one type there will be a choice of outlet proportions. It will also be evident that the tentative selection of an outlet having a wide spread may be unsatisfactory from the standpoint of throw, in which event a second choice should be made and the procedure repeated.

In the case of a heating problem, the method of solution is the same, but the manufacturer's data must, of course, be based on tests with air above room temperature.

TYPES OF SUPPLY OUTLETS

Grille, registers or outlet design for attaining uniform distribution and minimum air resistance consists of various fixed and adjustable arrangements. Some types are designed with directing air blades, fins, bars, louvres, or thin metal strips shaped into a series of grooves or tubes, all of which may be set into a suitable round, square or rectangular frame. In order to attain desired long or short air throws, the emergence of air from the outlet may be directed to straight, deflecting, converging or jet air streams depending upon the outlet design. Designs which direct the air stream to produce an ejector effect within the enclosed space tend to mix the room air with the conditioned air to provide uniform distribution.

Centrally located ceiling or wall type outlets arranged for completely diffusing the air consist of several round, hollow, cone-shaped flaring members placed in the proper relationship to each other. The velocity of emergence of the air from the unit can be made practically uniform over the entire surface of the outlet, and the velocity in any direction may be varied to any desired value by adjusting the position of the cones. One or more of the smaller flaring members act as ejectors and injectors which draw a small proportion of the room air into the air spreader where it mixes with the conditioned air before it is discharged.

An idea for producing even distribution of air consists of a perforated ceiling made of a suitable architectural surface and installed a small distance below the normal ceiling level of the room. In the space provided by this suspended ceiling a plenum chamber is formed into which the conditioned air is introduced. From the plenum space the air is permitted to diffuse through the large number of small ceiling openings into the room.

Railroad Cars

The early practice of air conditioning railroad passenger cars consisted of a system of bulkhead distribution for the conditioned air. The air was discharged through an inlet opening at each end of the car toward the middle with the flow parallel to the long dimension of the car. This type of installation resulted in drafts in the middle of the car and was considered unsatisfactory except for small sections that did not require large quantities of air. Later designs incorporated a duct delivery system on each side of the car roof directing the air through numerous inlet openings toward the middle of the car where the two air streams come in contact and deflect downward, gradually filtering into the aisle. At the present time, several center duct air distribution systems are used in railroad car applications. In some instances, square or circular ceiling outlets connected to a center duct have been used, which distribute the air along the ceiling in widening circles and at right angles to the inlet opening. Another method consists of a continuous slot in the bottom of the duct to which is attached a flat plate so that the air is deflected along the car ceiling. There are also installations in which the ceiling of the car is constructed of perforated metal through which the conditioned air flows through thousands of openings from a plenum chamber. Extensive tests of all methods of air distribution indicate that desirable results are obtained from an inside center duct with a large number of openings.

Sleeping cars present a special problem in air distribution on account of the berth curtains. In some cars, each lower berth is equipped with an individual fan to draw in cooled air from the aisle. In other cars, small individual ducts with adjustable air outlets deliver air from the central supply system to each lower berth. Upper berths require no special arrangement.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every outlet will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are listed:

1. Dampers on the supply grilles.
2. Dampers on the return grilles.
3. Dampers in the supply ducts.
4. Dampers in the return ducts.
5. Reducing the effective area of some outlets by blank-offs.
6. Combinations of dampers in both supply and return air.

Dampers on the supply grilles themselves are objectionable because of their effect on the air stream. Dampers on the return grilles are frequently helpful in building up a static pressure in the room to prevent infiltration of outside air, and at the same time reduce the volume of incoming air. However, it is frequently impossible to sufficiently reduce the incoming air by this method alone. A damper in the supply duct some distance back of the outlet forms a very satisfactory means of regulating the flow without disturbing distribution across the outlet face. A damper in the return air duct has the advantage over one immediately behind the grille in that it does not tend to create high localized velocities through the grille as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the outlet face can frequently be used satisfactorily, although determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the outlet, and ahead of the acoustic duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section*.

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Characteristics of Registers and Grilles, by J. H. Van Alsburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 245).

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PROBLEMS IN PRACTICE

1 ● What important factors are involved in the correct distribution of air to an enclosed space?

Not only is it important to distribute the air from the fan to the various spaces served by the system, but also the air must be properly distributed within the enclosed space to give complete satisfaction.

2 ● Upon what basis should the selection of supply outlets be based?

If possible, the selection should take advantage of the maximum velocity permissible from a noise standpoint.

3 ● What factors in grille design effect the length of air throw?

a. The temperature difference between incoming and room air, *b.* height of grille above floor, *c.* face velocity, *d.* core area, *e.* core aspect ratio, and *f.* design of grille.

4 ● How does the height of a supply outlet affect the temperature differential within a room?

In general, the higher the outlet is above the floor, the greater may be the difference between room air and incoming air temperatures.

5 ● Under conditions prevalent in a large room, how does the intensity of sound develop at an air outlet vary?

The intensity of sound energy is substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units in the room.

6 ● What are the essential differences between a high velocity long throw and short throw grille?

Generally, a high velocity long throw grille is used where a large compact mass of air is projected with a reduction in the periphery of the air stream whereas, with a short throw grille design the periphery of the air stream is expanded as much as possible to increase the scrubbing action between the incoming air stream and the stationary air.

7 ● What type of system is generally used in a large continuously operated theatre?

Most large continuously operated theatres are provided with a complete downward system of air distribution. With this system a large number of outlet openings are provided each of which discharges air in a thin horizontal stream at high velocity in order that the cool air would be mixed with the area in the theatre before it reaches the patrons. In this type of system the best distribution is obtained when a sufficient number of exhaust openings are located under the seats.

8 ● What means are available for balancing a system to secure the desired amount of air in each space?

Ways in which this may be accomplished are by: *a.* dampers on supply and return grilles, *b.* dampers in supply and return ducts, *c.* reduction of the effective area of some outlets by blank-offs, and *d.* combination of dampers in both supply and return air duct systems.

Chapter 29

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Proportioning the Losses, Sizes of Ducts, General Rules, Procedure for Duct Design, Air Velocities, Proportioning the Size for Friction, Main Trunk Ducts, Equal Friction Method, Duct Construction Details

THE flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_V be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60\sqrt{2g\,H_V}$$

The factor g is the acceleration due to gravity, or 32.16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_V}{\frac{h_V}{12}} = \frac{62.4}{d}$$

or

$$H_V = 5.2\,\frac{h_V}{d}$$

therefore,

$$V = 1096.5\sqrt{\frac{h_V}{d}} \qquad (1)$$

where

V = velocity in feet per minute.
h_V = velocity head or pressure in inches of water.
d = weight of air in pounds per cubic foot.

For standard air (70 F and 29.921 in. barometer) d = 0.07492 lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5\sqrt{\frac{h_V}{0.07492}} = 4005\sqrt{h_V} \qquad (2)$$

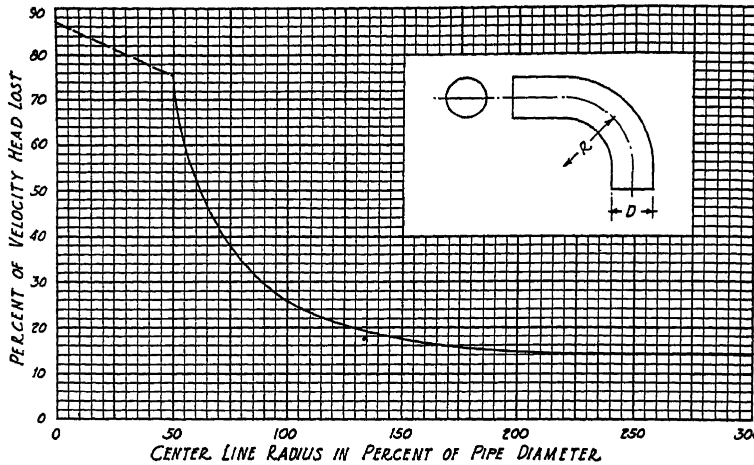

FIG. 1. CURVE SHOWING LOSS OF PRESSURE IN ROUND ELBOWS

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses are those due to the friction of the air against the sides of the duct. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Pressure Losses

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to

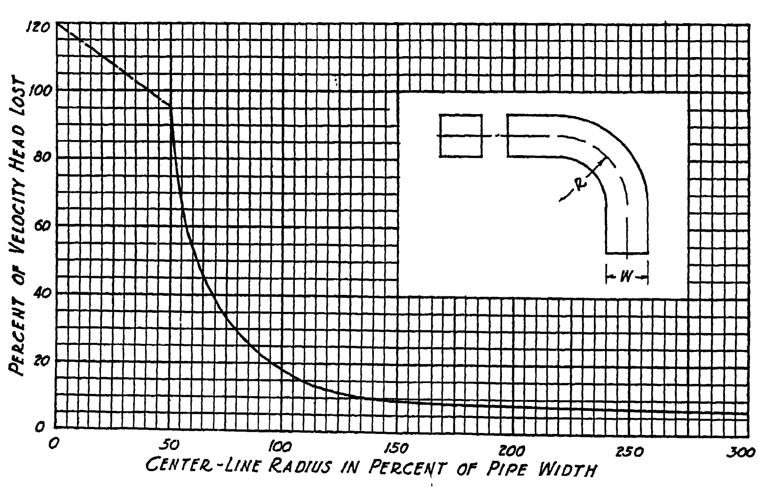

FIG. 2. CURVE SHOWING LOSS OF PRESSURE IN SQUARE ELBOWS

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design. It is customary to express dynamic losses in terms of the percentage of the velocity head; in other words, the percentage of that pressure corresponding to the average velocity in the duct which is expressed in terms of inches of water gage. Figs. 1 and 2 show the effect of changing the radius of elbows of square and rectangular section[1]. These charts are based on tests of pipe elbows of ordinary good sheet metal construction. For example, a five-piece round pipe elbow having a centerline radius of one diameter has a loss of about 25 per cent of the velocity head. At a velocity of 2000 fpm the corresponding head is 0.25 in. water gage, and at this velocity the elbow just referred to would cause a pressure drop of 0.063 in. water gage. Experience has shown that good results may be obtained when the radius to the center of the elbow is 1½ times the pipe diameter. The pressure drop will then be approximately 17 per cent of the velocity head for round ducts, and 9 per cent for square ducts. Very little advantage is gained in making elbows with a radius of more than two diameters[2].

Friction Losses

Friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter. Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and standard air (70 F and 29.921 in. barometer)

$$h_L = f\frac{L}{D}h_V = \frac{L}{CD}\left(\frac{V}{4005}\right)^2 \tag{3}$$

For rectangular ducts

$$h_L = fL\left(\frac{a+b}{2ab}\right)h_V = \frac{L}{C}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4005}\right)^2 \tag{4}$$

where

h_L = loss of head, inches of water.

$h_V = \left(\frac{V}{4005}\right)^2$ = velocity head, inches of water.

V = velocity of air, feet per minute.

L = length of pipe, D = diameter of pipe, a, b = sides of rectangular duct — all in feet.

f = coefficient of friction.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss.

For all practical purposes C varies only with the nature of the pipe surface: C = 60 for perfectly smooth pipe; = 55 for pipe as used in planning

[1]Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 366).

[2]Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, p. 365, August, p. 427, September, p. 483, 1936).

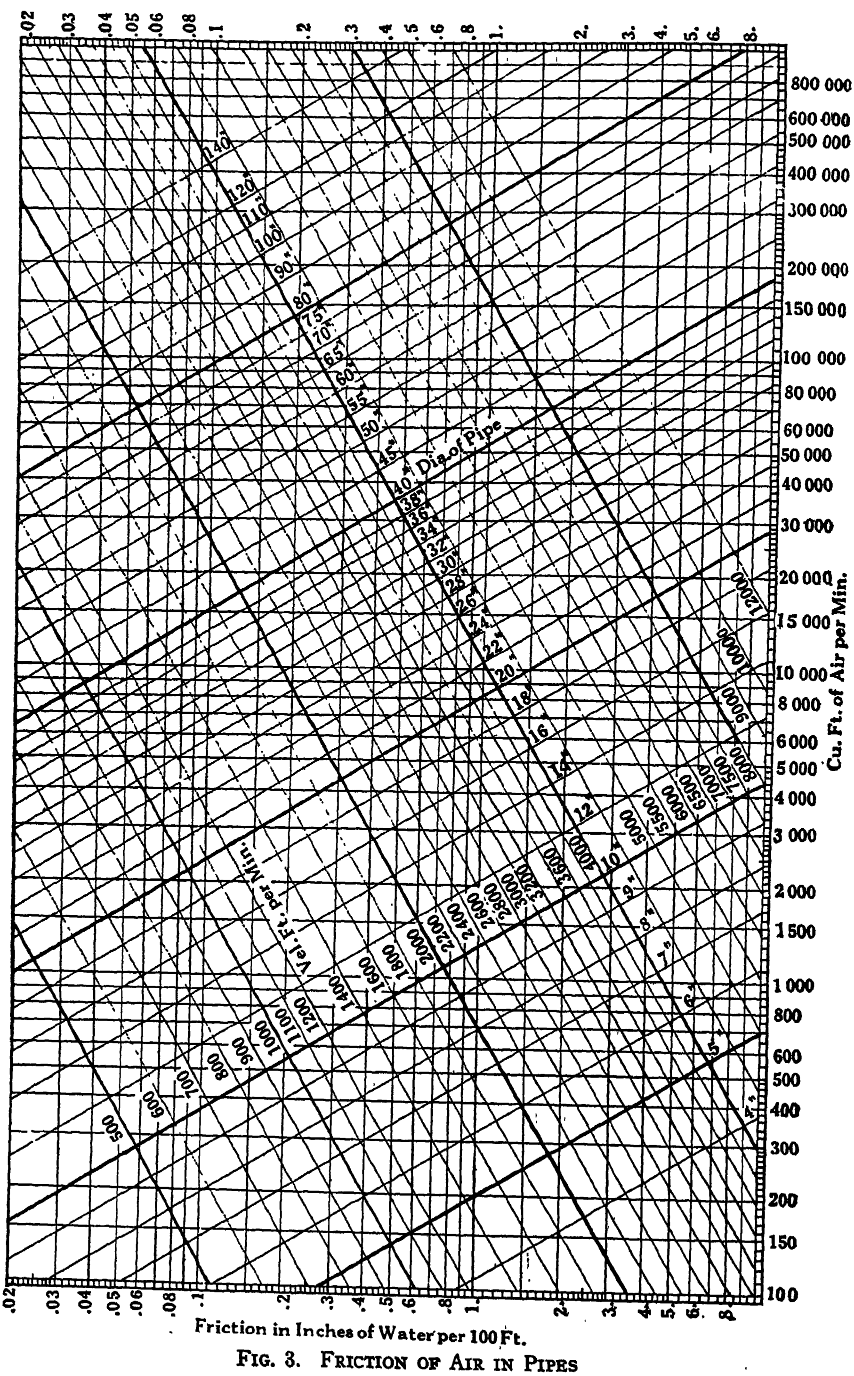

FIG. 3. FRICTION OF AIR IN PIPES

mill exhaust systems; = 50 for heating and ventilating ducts; = 45 for smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzsche states (and numerous tests check very closely) that f varies inversely as the 2/7 power of the pipe diameter, and inversely as the 1/7 power of the velocity, or inversely as the 1/7 power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24-in. galvanized swedged pipe:

$$h_L = 1.1 \frac{L}{CD^{9/7}} \left(\frac{V}{4005}\right)^{13/7} \tag{5}$$

The preceding formulae are based on standard air, and for other conditions the friction varies directly as the air density and inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

Fig. 3 is a convenient chart for determining the friction loss for various air quantities in ducts of different sizes. The general form of this chart is familiar, but it should be noted that it is corrected for changes in the coefficient of friction based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the velocity. Fig. 3 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of 24-in. round galvanized-iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C, the friction loss as read from the chart should be multiplied by $\frac{50}{40}$.

Example 1. Assume that it is desired to pass 10,000 cfm of air through 75 ft of 24-in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 3 and move horizontally left to the diagonal line marked 24 in The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.59 in.; then for 75 ft the friction will be $0.75 \times 0.59 = 0.44$ in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Proportioning the Losses

Other losses of pressure occur at the entrance to the duct, through the heating units, and at the air washer. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses ⅓ to ½ and the loss through the heating units at less than ½ of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

SIZES OF DUCTS

The sizes of ducts and flues for gravity or mechanical circulation of air are usually based on the losses due to friction, and these losses must be

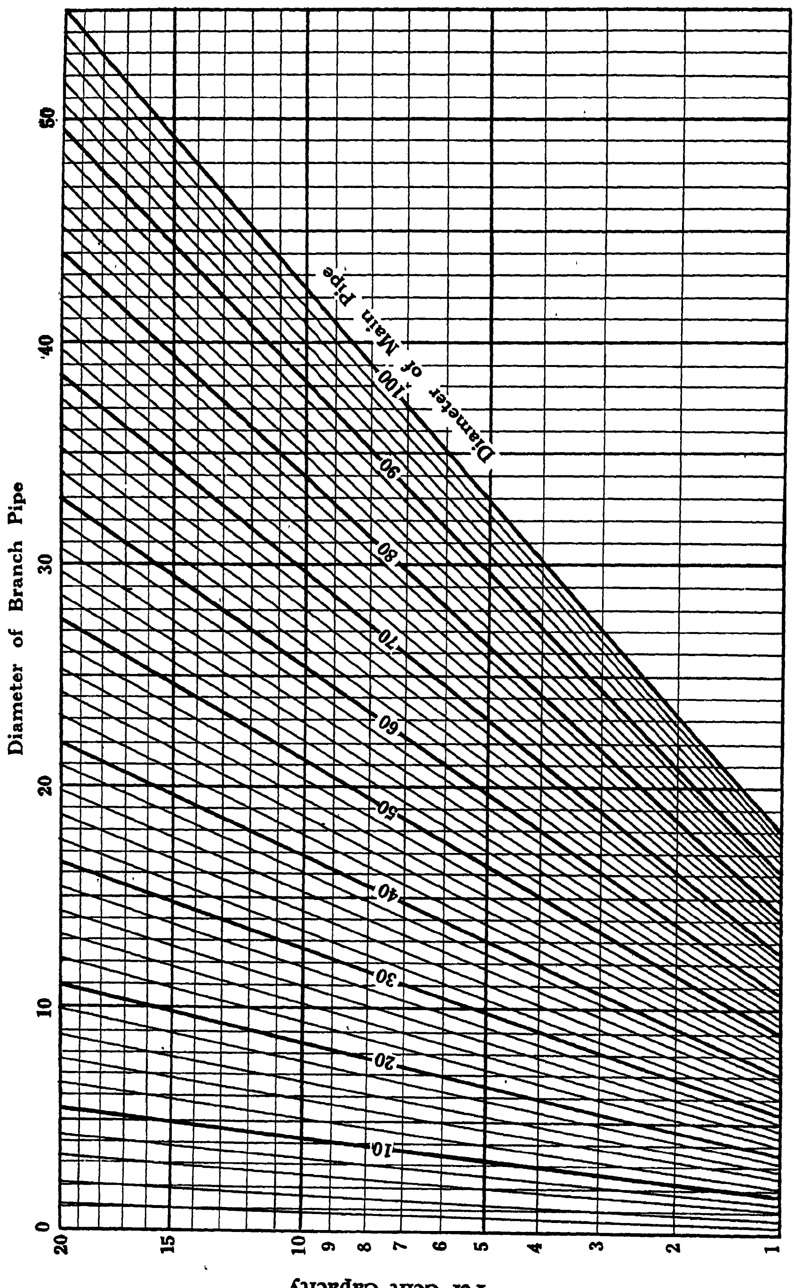

FIG. 4. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH (1 TO 20 PER CENT CAPACITY)

kept within the available pressure difference. This pressure difference in mechanical ventilation is that derived from the fan, while in gravity ventilation the aspirating effect due to the temperature and height of the column of heated air causes the pressure difference.

General Rules

The general rules to be followed in the design of a duct system are:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided.
3. The sides of all ducts or flues should be as nearly equal as possible. (In no case should the ratio between long and short sides be greater than 10 to 1.)

Procedure for Duct Design

The general procedure for designing a duct system is as follows:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of heat.
3. Divide the building into zones and proportion the volume of air necessary to supply the heat for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:
    a. *Velocity Method.* Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections.
    b. *Friction Pressure Loss Method.* Proportion the duct for equal friction pressure loss per foot of length.
6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The following velocities of air are considered standard for public buildings:

1. Through the outside air intakes, 1000 fpm.
2. Through connections to and from heating unit, 1000 to 1200 fpm.
3. Through the main discharge duct, from 1200 to 1600 fpm.
4. In branch ducts, 600 to 1000, and in vertical flues, 400 to 800 fpm.
5. In registers or grilles, 200 to 400 fpm depending upon the size and location. If diffusers of proper design are used, 25 per cent higher air velocities are permissible.

These duct velocities may safely be increased 20 per cent if first class construction is used to prevent any breathing, buckling, or vibration. High velocities at one point in the system neutralize the effect of proper design at all other points; hence the importance of splitters in elbows and similar precautions. For industrial buildings noise is seldom considered, and main duct velocities as high as 2800 or 3000 fpm may be used where conditions will permit. For department stores and similar buildings,

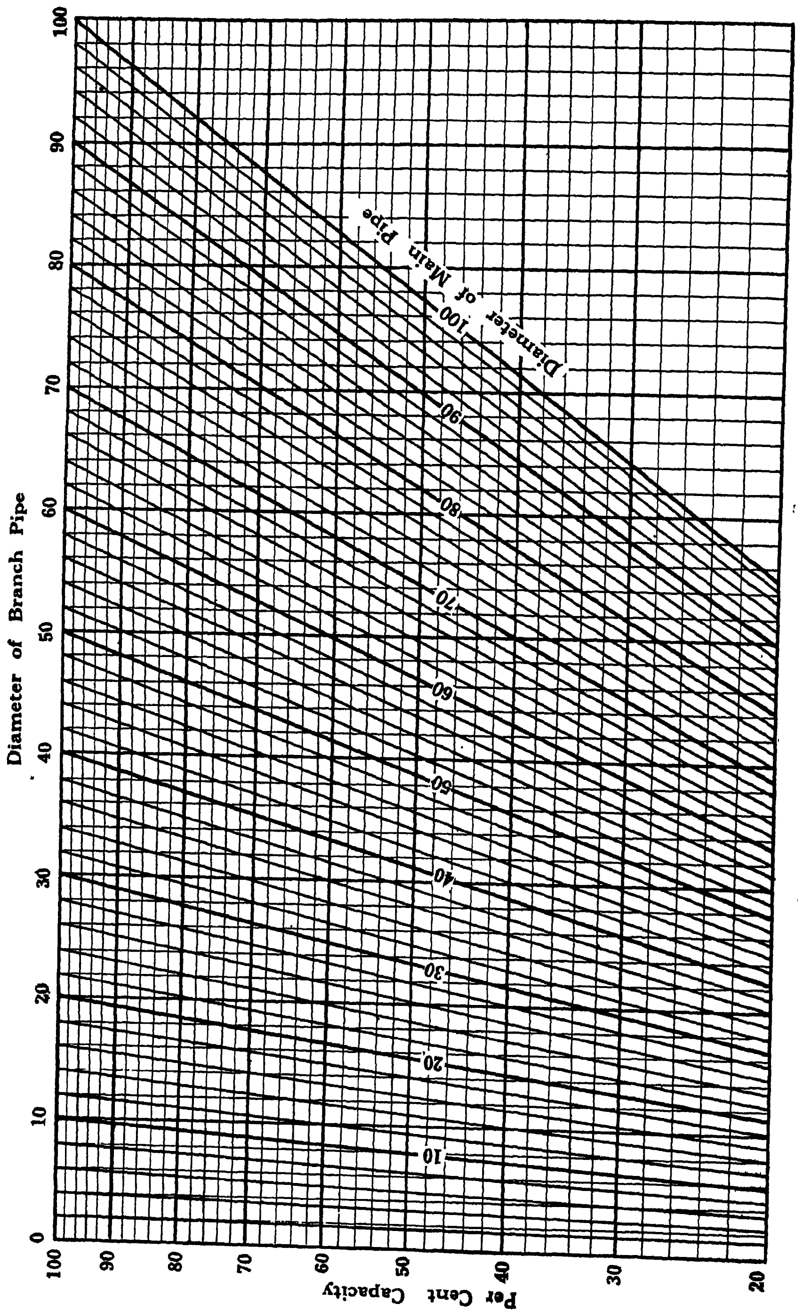

FIG. 5. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH (20 TO 100 PER CENT CAPACITY)

maximum velocities with good construction and design may be as high as 2000 or 2200 fpm in main ducts, with suitable reduction in branches and outlets. With these velocities first-class duct construction is essential.

Proportioning the Size for Friction

By means of Figs. 4 and 5 the diameter of branch pipes necessary to carry a given percentage of the total air in the main pipe and to maintain equal friction per foot of the length through the entire system may be determined. These charts, as well as Fig. 3, are based on the assumption that the coefficient of friction varies inversely as the 1/7 power of the capacity.

Example 2. Suppose a 60-in. main pipe is to be used, and it is desired to know the size of branch pipe required to carry 50 per cent of the total air in the main. Find 50 per cent at the left of the chart, move right to the 60-in. diagonal line and note directly above at the top of the chart that the branch pipe will be 46.5 in. in diameter.

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalents of rectangular ducts for equal friction and capacity, which are based on values determined from Formula 6:

$$d = 1.265 \sqrt[5]{\frac{(a\,b)^3}{a + b}} \tag{6}$$

where

a = one side of rectangular pipe, feet or inches.
b = other side of rectangular pipe, feet or inches.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, feet or inches.

To obtain the size of rectangular ducts for different capacities, but of the same friction per foot of length, first obtain the equivalent round pipe for equal friction. Thus, if a branch of sufficient size to carry 30 per cent of a 12 x 36-in. pipe is desired, it is found from Table 1 that the main is equivalent to a 22.2-in. diameter round pipe. From Fig. 5, 30 per cent of this is a pipe 14.3 in. in diameter, and referring again to Table 1, the rectangular equivalent branch is a 12 x 14 in., 10 x 17¼ in., or any other desirable combination.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is required, it will be twice that of a 40 x 12-in. duct, or 2 × 23.3 = 46.6 in.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION[a]—(Continued)

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

[a]Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3; 5 × 7 = 6.5.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Concluded)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48	Side Rectangular Duct	50	54	60	66	72	78	84	88
26	28.6												50	55.0							
28	29.7	30.8											52	56.1							
30	30.7	31.9	33.0										54	57.2	59.4						
32	31.7	32.9	34.1	35.2									56	58.3	60.5						
34	32.7	33.9	35.1	36.3	37.4								58	59.3	61.6						
36	33.7	34.9	36.1	37.3	38.5	39.6							60	60.3	62.7	66.0					
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8						62	61.3	63.7	67.1					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0					64	62.2	64.7	68.2					
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2				66	63.2	65.7	69.3	72.6				
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4			68	64.1	66.6	70.3	73.7				
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6		70	65.0	67.6	71.3	74.8				
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8	72	65.9	68.5	72.3	75.9	79.2			
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0	74	66.8	69.4	73.3	76.9	80.3			
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0	76	67.6	70.3	74.2	77.9	81.4			
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0	78	68.4	71.2	75.2	78.9	82.5	85.8		
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0	80	69.2	72.1	76.1	79.9	83.6	86.9		
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0	82	70.1	73.0	77.1	80.9	84.6	88.0		
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9	84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7	86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6	88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6	90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6	92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5	94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5	96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

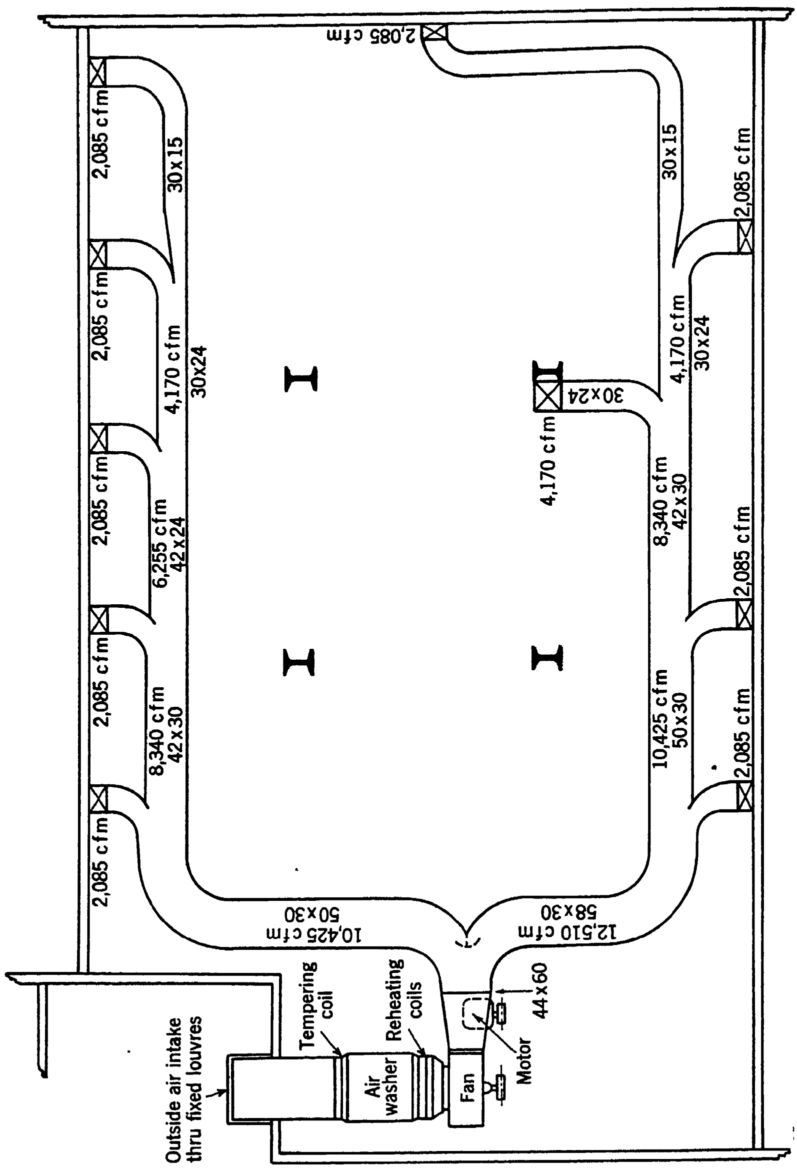

FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

MAIN TRUNK DUCTS

A main duct with branches is generally used to convey tempered air for ventilation purposes only. In place of individual ducts, a comparatively large main duct supplies air by branches to the room or rooms. The velocities vary according to the nature of the installation and the degree of quietness required. At the start of the run a velocity as high as 2000 fpm may be used, but this is considered the maximum for public building work, and is reduced to from 400 to 800 fpm in the risers. This duct system may be designed so that the loss of pressure in the branches is equalized in a manner similar to that previously described.

Equal Friction Method

Example 3. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices.

The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-minute air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The clear area of the fresh air inlet is based on a velocity of 1000 fpm or $\frac{22,935}{1000} =$ 22.94 sq ft. If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F. The washer with its automatic control will then raise the temperature from 32 F to 42 F. If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation. The reheating coil is selected to raise the temperature of the air from that leaving the air washer to 70 F. The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft. For more detailed information on tempering coil and air washer control, see Chapter 37.

Since the plan shows a moderately short run of main duct with no risers near the fan outlet, a fan should be selected which will have the required capacity of 22,935 cfm with a maximum velocity through the fan outlet of 1400 fpm. The outlet area, therefore, should be 16½ sq ft.

The main pipe size should be selected to give a velocity equal to or less than the velocity at the fan outlet. Choosing a 56-in. pipe with a cross-sectional area of 17.1 sq ft, the velocity in the main pipe will be 1340 fpm. Using the friction pressure loss method this 56-in. main pipe will be taken as the basis of calculation.

Fig. 6 shows the amount of air to be handled by each section of pipe. Expressing the volume handled by each section as a percentage of the total volume and using the charts, Figs. 4 and 5, the pipe sizes are as shown in Table 2.

TABLE 2. PIPE SIZES FOR EXAMPLE 3[a]

Volume of Air (cfm)	Per Cent of Total Volume	Diameter of Pipe (Inches)	Equivalent Size of Rectangular Duct (Inches)
22,935	100.0	56	60 x 44
12,510	54.6	45	58 x 30
10,425	45.4	42	50 x 30
8,340	36.3	39	42 x 30
6,255	27.2	35	42 x 24
4,170	18.2	29½	30 x 24
2,085	9.1	23	30 x 15

[a]Velocity through diffusers (not shown) to be approximately 300 fpm.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 in. × 30 in. and 50 in. × 30 in. branches.

The rectangular equivalents are selected from Table 1; the width to depth proportion will be determined by construction requirements and ease of fabrication. The calculation of the friction is as follows:

The longest run from the fan outlet to diffuser is 150 ft 0 in.; 150 ft of 56-in. pipe is equivalent to $\frac{150 \times 12}{56}$ .... 32.2 dia.

Two 45-in., 90-deg elbows $(2 \times \frac{45}{56} \times 8.5)$ .... 13.7 dia.

(Assume each elbow equivalent to 8.5 diameters of duct, Fig. 1.)

Two 23-in., 90-deg elbows $(2 \times \frac{23}{56} \times 8.5)$ .... 7.0 dia.

Two 23-in., 90-deg elbows in riser $(2 \times \frac{23}{56} \times 30)$ .... 24.7 dia.

(Two bad elbows in riser, each equivalent to 30 diameters of duct).

Total diameter of 56-in. pipe 77.6

The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{77.6}{50} \times 0.112 = 0.174$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss. This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0.027 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in. The regain equals ⅔ (0.122 − 0.027) = 0.063 in.

The net static pressure loss in the duct is:

0.174 in. − 0.063 in. .... 0.111 in.

Other friction losses are as follows:

(1) Fresh air intake 1000-fpm velocity (1½ heads × 0.0625) .... 0.094 in.
(2) Tempering coil loss (from manufacturer's tables) .... 0.100 in.
(3) Air washer loss (from manufacturer's tables) .... 0.250 in.
(4) Reheating coil loss (from manufacturer's tables) .... 0.100 in.
(5) Allowance for regulating dampers and diffusers .... 0.100 in.

Static pressure loss of system .... 0.755 in.

The fan should be selected from the manufacturer's ratings which, according to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[1], will deliver 22,935 cfm at a static pressure of 0.755 in. and which has an outlet area of 16½ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves

[1]See Chapters 27 and 45.

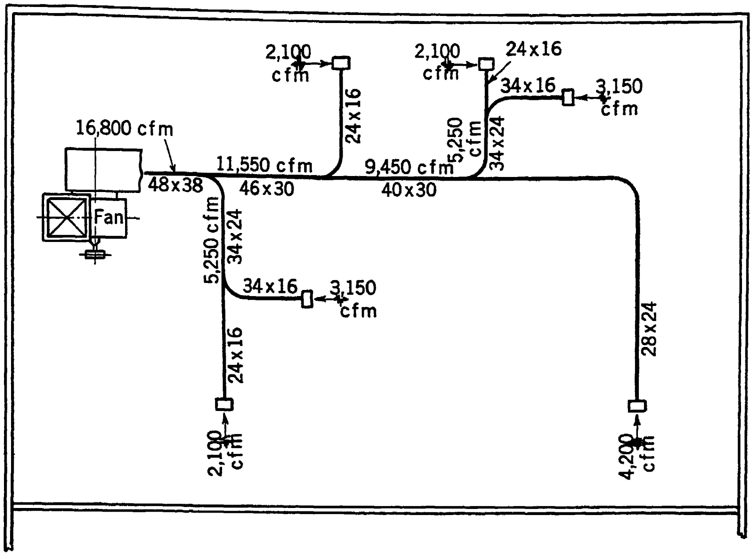

FIG. 7. EXHAUST SYSTEM LAYOUT

the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers and supply branches in order that equalization of air flow can be attained.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe. Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3 for both round and rectangular ducts.

TABLE 3. PIPE SIZES FOR EXAMPLE 4[a]

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
.4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21.6	16 x 24

[a]Velocity through intake grilles (not shown) to be approximately 400 fpm.

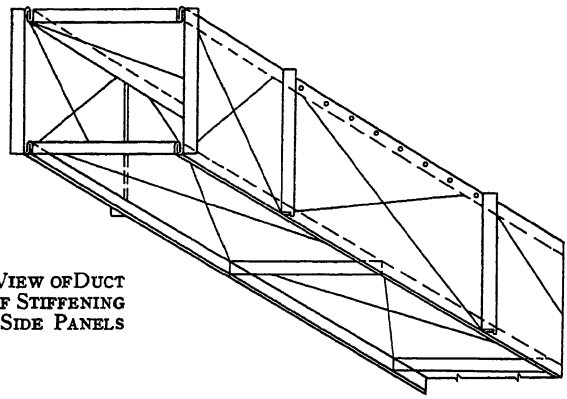

FIG. 8. ISOMETRIC VIEW OF DUCT SHOWING LOCATION OF STIFFENING SEAMS ON TOP AND SIDE PANELS OF DUCT

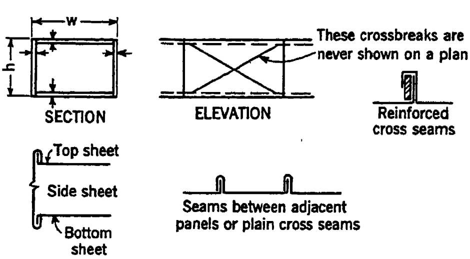

FIG. 9. DETAILS OF SEAMS

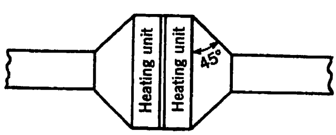

FIG. 10. METHOD OF INSTALLING HEATING UNIT

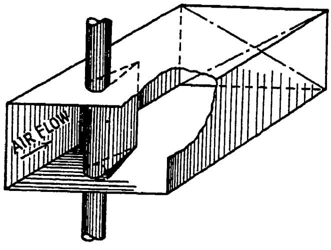

FIG. 11. INSTALLATION OF EASEMENT IN DUCT AROUND OBSTRUCTION

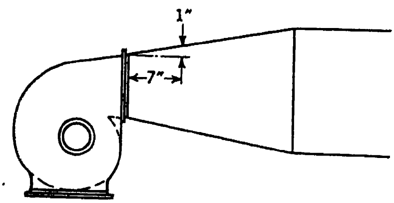

FIG. 12. FAN DISCHARGE CONNECTION

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft.

(1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ .... 25.6 dia.

Two 28½-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ .... 36.4 dia.

(Two bad elbows in riser each equivalent to 30 diameters of duct).

One 28½-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 8.5}{47}\right)$ .... 5.2 dia.

Total diameter of 47-in. pipe .... 67.2 dia.

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 0.122$ in.

Taking 50 diameters as one head loss, then $\frac{67.2 \times 0.122}{50}$ .... 0.164 in.

(2) Intake loss from grille (1½ heads at a 400 fpm velocity 1½ × 0.01) .... 0.015 in.

(3) Static pressure required to produce one velocity head at 1400 fpm .... 0.122 in.

(4) Loss occasioned by step-up of velocity (0.20 × 0.122) .... 0.024 in.

(This loss varies from 0.05 to 0.40 velocity head depending upon the nature of the change. For average systems 0.20 velocity head is a close approximation.)

Static pressure loss on inlet side .... 0.325 in.

To this must be added the resistance on the discharge side of the fan. A fan outlet velocity of approximately 1500 to 1600 fpm may be used. Assuming the fan outlet to be equivalent in area to a 45-in. pipe, the velocity is 1525 fpm.

Loss on discharge (15 ft from fan outlet to discharge):

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in. pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0.145 and the discharge-side loss is $\frac{0.145 \times 4}{50} = 0.012$ in. The total static pressure loss of the system is then:

$$0.012 + 0.325 = 0.337 \text{ in.}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0.337 in. and to have an outlet velocity of 1525 fpm. Outlet area 11 sq ft.

Where there are one or more ducts with branches, the velocity of air in the ducts may be either chosen arbitrarily or calculated for friction losses. When arbitrary values are assigned, a certain amount of dampering should be provided for; this will be small when the method chosen permits a drop in velocity as the quantity of air is reduced.

After the total air quantity and the size of fan are ascertained, the main duct is usually fixed as being at least equal in area to the fan outlet, or perhaps 10 per cent greater. From this main pipe all others are proportioned. For example, if the main duct is 30 in. in diameter, a branch to carry 10 per cent of the total capacity should be 12.7 in. in diameter (see Fig. 4) in order to have the same friction per foot of length, while one

carrying one-half the total capacity of a 30-in. main with the same friction loss per foot would be 23.4 in. in diameter. By this method of equalizing friction it is unnecessary to consider the resistance of each section of pipe independently, but only to know the distance from the fan outlet to the end of the longest run of pipe, the number and size of elbows, and the diameter and velocity in the largest pipe.

Example 5. If the greatest length of piping in a system is 130 ft with a 26-in. diameter main pipe and one 20-in. elbow, the piping having been designed for equal friction per foot of length, the friction would be the same as for 130 linear feet of 26-in. pipe, or 60 diameters. To this should be added the friction loss in elbows, in this case one 20-in. elbow, which has a loss equivalent to 8.5 diameters of 20-in. pipe. This in turn is $\frac{20}{26} \times 8.5 = 6.6$ diameters of 26-in. pipe. The total equivalent length of the system will then be $60 + 6.6$, or 66.6 diameters. Since 50 diameters is equivalent to one velocity head, the loss is $\frac{66.6}{50} = 1.33$ times the velocity head. If the velocity is, for example, 2200 fpm, corresponding to 0.3-in. pressure, the friction loss of the system will be $1.33 \times 0.3 = 0.399$ in.

TABLE 4. SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[a]

GAGE	WIDTH OF DUCT	SEAM	REINFORCED SEAM
26	Up to 12 in.		
24	13 in. to 30 in.	1	
22	31 in. to 48 in.	1	
22	49 in. to 60 in.	1½	⅛ in. x 1⅜ in.
20	61 in. to 90 in.	1½	⅛ in. x 1⅜ in.

[a]If panels are not cross-broken two gages heavier material should be used.

Frequently the prevention of sound in a heating or ventilating system imposes more severe restrictions than the prevention of excessive pressure drop. This question is highly involved and requires consideration of many factors. The air velocities to be used will vary with the standard of construction used in the ducts themselves as well as with the nature of the occupancy and the construction of the building. In general, architects and engineers who leave the details of duct construction to the contractor must, of necessity, design for lower velocities than might be required for quiet operation if proper construction details were always followed. The contractor may be expected to build the ducts by the least expensive methods, and the engineer must anticipate this. For further information on noise reduction, see Chapter 30.

DUCT CONSTRUCTION DETAILS

If panel construction is used with standing seams or similar reinforcement, and the panels are cross-broken to give rigidity, there is less likelihood of vibration due to air flow, or deflection due to air pressure. Elbows made without splitters, and improperly shaped transformation sections produce high local velocities which are the cause of noise in duct work. The use of first-class duct construction with well-designed transformation sections and splitters in elbows tends to maintain relatively uniform velocities with decrease in turbulence and in the noise produced.

Figs. 8 to 12 show acceptable construction details for rectangular ducts, elbows, and transformation pieces or connections. Other methods are also acceptable, such as the use of angle iron stiffeners for large ducts. Good construction is essential to the elimination of duct noises and for the prevention of a flimsy installation.

Fig. 8 is an isometric view of a duct showing the location of the stiffening seams on the top and side panels. The cross seams should not occur at the same place but should be staggered as indicated. Heating units should be installed as shown in Fig. 10 with the duct connections making an angle of not less than 45 deg, but preferably 60 deg. Fan discharge connections should have a maximum slope of 1 in 7, as indicated in Fig. 12. Whenever a pipe or other obstruction passes through a duct an easement should be placed around the pipe as indicated in Fig. 11. The recommended gages for rectangular sheet metal duct construction are given in Table 4.

REFERENCES

Fan Engineering, Buffalo Forge Co.

Heat Power Engineering, by Barnard, Ellenwood, and Hirshfeld, Part III.

Mechanical Engineers' Handbook, by Lionel S. Marks, McGraw-Hill Book Co.

The Flow of Liquids, by W. H. McAdams (*Refrigerating Engineering*, February, 1925, p. 279).

A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler (*A.S.M.E. Transactions*, Hydraulics Section, August 31, 1933, p. 7).

PROBLEMS IN PRACTICE

1 ● Determine the equivalent number of diameters of straight pipe equivalent to a 90 deg elbow having center line radii of (a) 100 per cent, (b) 150 per cent, abd (c) 200 per cent of the pipe diameter.

Assume 1 velocity head lost in 50 diameters.

From Fig. 1 the per cent of velocity head lost:

a. For 100 per cent radius is 25.5 per cent $\times$ 50 = 12.8 diameters straight pipe.
b. For 150 per cent radius is 17.0 per cent $\times$ 50 = 8.5 diameters straight pipe.
c. For 200 per cent radius is 14.5 per cent $\times$ 50 = 7.3 diameters straight pipe.

2 ● Why is it desirable to make elbows with a radius equal to one and one-half times the pipe diameter?

Reference to Figs. 1 and 2 will show that while the loss of velocity head, as indicated by the curves, shows considerable variation for elbows between the range of 50 and 150 per cent radius, the line is practically straight after 150 per cent, indicating very little variation in loss of head for elbows of larger radius.

3 ● What is the best shape to use for ducts?

The shapes to be used in designing ducts, in the order of their preference, are round, square, and rectangular.

4 ● What determines which shape to use?

Structural and space conditions. Because ducts are as a rule part of the building or structure, it is necessary to proportion their sizes to fit the spaces available.

5 ● What is meant by "arbitrarily fix the velocity in the various sections?"

When using the velocity method as a basis for design, the maximum allowable velocity is fixed for the main supply duct at the fan, and this velocity is gradually decreased as each branch or outlet is taken off the main supply duct.

6 ● Which system of duct design is to be preferred, the velocity method or the friction pressure loss method?

The friction pressure loss method can be used to advantage where no structural or building conditions limit the shape of the ducts. Where these limiting conditions exist the velocity method is to be preferred.

7 ● Are the grille sizes figured on the same basis as the outlets?

The free area through the grilles is figured the same as the outlets, and this area is increased from 20 to 50 per cent, depending on the design of the grille, to allow for the loss of area caused by the construction of the face of the grille.

8 ● Where it is necessary to provide steel angle braces, how far apart should they be spaced?

Angle braces for large ducts should be placed on 3-ft 0-in. centers.

9 ● How much air will a 10-in. by 24-in. duct handle if it is part of a system designed on a pressure drop of 0.1 in. per 100 feet of run?

1450 cfm (Table 1 and Fig. 3).

10 ● How does a splitter at a duct junction influence the volume of the air going through each branch?

A splitter facing the direction of air flow cuts off the air and delivers the desired amount to the branch.

11 ● Why does a wide, shallow duct offer more resistance to the flow of air than does a square duct of equal cross-sectional area?

The perimeter of the wide, flat duct is greater than that of the square-section duct, so the former has the greater frictional area which increases the resistance and thus reduces the volume at any given pressure.

12 ● What methods are used to keep large ducts from vibrating because of air pulsations, and from sagging because of their own weight?

External bracing, such as standing seams, or structural shapes, like tees or angles, should be placed across the top and bottom. Exterior braces or cross buckling of metal sheets in diagonal panels may be used for the sides of large ducts.

13 ● What velocities of air flow should be used in the trunk ducts of a ventilating system in a public building?

From 1200 to 1600 fpm.

14 ● In a ventilating system in a residence, what is the recommended air velocity through supply registers and grilles?

400 fpm.

Chapter 30

SOUND CONTROL

Decibel Defined, Apparatus for Measuring Noise, Problem of Sound Control, Acceptable Noise Levels, Controlling Vibration from Machine Mountings, Controlling Noise through Room Wall Surfaces, Noise Transmitted Through Ducts, Duct Lining Factor

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is not assumed that the ventilating and air conditioning engineer will attempt to improve the acoustics of the space that is being conditioned, but the designer should have at least enough fundamental knowledge of the acoustical effects of the system which is being designed to be sure that no damaging effects occur to the existing acoustical properties. It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

In the United States and England the unit of noise measurement is the *decibel* (db). In Germany this unit is called the *phon.* The decibel is defined by the relation $N = 10 \log \frac{I_1}{I_0}$, where N is the number of decibels by which the intensity flux I_1 exceeds the intensity flux I_0. The intensity flux is the measure of the energy contained in a sound wave and is defined in terms of microwatts per square centimeter of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitrary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose the threshold of audibility for the average human ear at a frequency of 1000 cycles per second has been selected. This reference threshold is 10^{-16} watts per square centimeter or 10^{-10} microwatts per square centimeter. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound

measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in tentative standards[1] published by the *American Standards Association.*

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying sensitivity of the ear at different frequencies in designing noise measuring equipment. The most satisfactory method of measuring noise is by means of a sound level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the threshold of audibility and usually contains a weighing network to make it less sensitive at those frequencies where the ear is less sensitive. For complete specifications relative to the approved type of sound level meters refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the function of the ventilating and air conditioning engineer is to add no acoustical hazard to the conditions already present in the room or building and the problem can be stated as:

a. To determine the noise level existing without the equipment.

b. To ascertain the noise level which would exist if the equipment were installed without sound control.

c. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in (*a*).

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.
2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.
3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have information available to deal with noises which may enter the room due to openings made into it to accommodate the equipment, such as cross talk between rooms connected with common ducts and noise transmitted to portions of duct system outside the conditioned space and through to its interior.

While the general problem may be logically outlined and the items of

[1]American Tentative Standards for Noise Measurement, *American Standards Association.*

[2]American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, *American Standards Association.*

knowledge necessary to its solution can be listed, the available information at present is lacking in certain respects. However, attention may be directed to that information which is currently available, and to furthermore outline a solution of the noise problem based on these data.

ACCEPTABLE NOISE LEVELS

Measurements of noise levels have been observed by several investigators in various rooms and locations. The information compiled in Table 1 is based on these data, which represent the best opinion on the

TABLE 1. TYPICAL NOISE LEVELS

Rooms	Noise Level in Decibels to be Anticipated		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc.	25	35	40
Theatres, Legitimate	25	30	35
Theatres, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc.	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	25	33	40
Private Offices, Acoustically Untreated	35	45	50
General Offices	45	55	60
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Building, Court Houses, Post Offices, etc.	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	60	70	80
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[a]	70	80
Pullman Car	55[a]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

subject now available. All levels are given in decibels above a reference threshold of 10^{-16} watts (corresponding to a pressure of 0.0002—dynes per square centimeter). Minimum, representative, and maximum levels are given for each application. These values are intended to indicate the variation which may be expected in different locations of the same type, but not the time variation which may be expected in each location.

The values shown in Table 1 are typical of those found currently in

existing spaces. They are, however, the noise levels of the room and not the noise levels of the ventilating or air conditioning equipment. If the noise level at the room of the equipment is kept at the levels shown in the table the equipment will not add to the acoustical hazard existing without it, provided the equipment noise is heard alone, but if both are heard together the total noise level in the room will be increased about 3 db. This is usually considered an acceptable result.

In some cases it is desirable to keep the equipment noise level at the room at such a value that it actually will not increase the noise level in the room to any measurable degree. This can usually be accomplished if the equipment noise at the room can be kept 10 db below the noise level shown in the table.

NOISE CREATED BY EQUIPMENT

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products.

Absence of this information makes it necessary to resort to indirect means in solving certain problems and also prevents a direct logical solution.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source and places the emphasis on ascertaining the level at the point where the sound enters the room rather than on its point of origin.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two kinds depending on how it reaches the room as:

1. Noise transmitted through the building construction.
2. Noise transmitted through the ducts.

It is convenient to further subdivide these two methods of delivery as:

1. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.
2. Noise transmitted through the ducts.
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From air current, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system.

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE THROUGH BUILDING CONSTRUCTION

It is impossible to select ventilating equipment which will operate without producing some mechanical noise, and since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building itself to such a degree as to make noisy conditions in the rooms which are to be air conditioned. Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the contact between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if it is improperly engineered, may actually increase the contact between the equipment and the floor upon which it is supported. In general, the flexible material should be loaded as heavily as possible without impairing its load-carrying capacity.

Controlling Vibration from Machine Mountings

The theory of the insulation of vibration was first worked out by Soderberg[3]. If a machine of mass m be supported by an elastic pad the amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by the following equation:

$$\tau' = \sqrt{\frac{r^2 + \dfrac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi n m - \dfrac{1}{2\pi n c}\right)^2}} \tag{1}$$

where

τ' = the so-called *transmissibility* of the support.
c = the compliance (that is, the reciprocal of the force constant).
r = the mechanical resistance owing to the viscous forces within the support.
n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.
m = the mass of the machine to be insulated.

It should be noted that not only must vibrations within the audible range of frequencies be considered, but those in the sub-audible range as well, since these may cause objectionable vibrations. All the possible frequencies should be considered in the calculation. Sometimes beat effects are introduced by slight irregularities of belts or pulleys that have much lower frequencies than those of the rotating elements.

[3]C. R. Soderberg, *The Electric Journal* (January, 1924), and succeeding articles. See also V. O. Knudsen, *Physical Review*, Vol. 32, 1928, p. 324, and A. L. Kimball, Journal *Acoustical Society of America*, Vol. 2, 1930, p. 297.

If r, the mechanical resistance, is very small, formula 1 may be written

$$\tau = \frac{1}{\frac{n^2}{n_0^2} - 1} \qquad (2)$$

where n_0 is the natural frequency of the machine upon the elastic pad,

$$n_0 = \frac{1}{2\pi}\sqrt{\frac{1}{mc}}.$$

In most cases of design of resilient machine mounting the effect of frictional resistance is small, and Equation 2 may be used. In such cases it is only necessary to know the natural frequency of the elastic pad or platform used under the desired loading and the transmissibility for any vibrational frequency of the machine may be obtained. However, this formula gives the theoretical maximum insulation which may be obtained and should be used with a liberal factor of safety. (A factor of 2 is common practice.)

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ^1 must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 50 vibrations per second, and for best results preferably 20.

When the forced frequency is low, it is frequently impossible to insulate for the fundamental forced frequency due to connecting pipe work and other relevant factors. In cases of this kind an effective installation of sound insulation may be obtained with a mounting which functions far above the fundamental forced frequency. For example, a compressor operating at 500 rpm has a forced frequency of 8.3 vibrations per second. By designing a mounting having a natural frequency of 20 to 25 vibrations per second, it is possible to isolate practically all of the noise.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support will be worse than nothing. In general, as Equation 1 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies

above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approxi-

TABLE 2. COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS[a]

The compliances and resistances given in the table are for specimens 1 in. thick and 1 sq cm in cross-section

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1.10 lb per board foot	12	0.25×10^{-6}	0.15×10^{5}
Corkboard	0.70 lb per board foot	8	0.50×10^{-6}	0.25×10^{5}
Fiber Board	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^{5}
Fiber Board	Carpet lining	10	0.40×10^{-6}	----------
Fiber Board	Insulating board	12	0.18×10^{-6}	----------
Fiber Board	Insulating board	15	0.16×10^{-6}	----------
Fiber Board	Insulating board	15	0.12×10^{-6}	----------
Anti-Vibro-Block	----------	5	0.60×10^{-6}	1.5×10^{5}
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0×10^{-6}	----------
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2×10^{-6}	----------

[a]From *Architectural Acoustics*, by V. O. Knudsen, p. 278.

mately because the resistance will be inversely proportional to the thickness and directly proportional to the area of cross-section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 1, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m. For the routine calculations in practice, r may be neglected with only a slight sacrifice of accuracy. Table 2 gives the values of c and r for a number of commonly used flexible materials.

Example 1. A machine weighing 1000 lb has a base area of 20 sq ft. Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in. slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than ⅓ lb per square inch. (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^{5}.)

The *transmissibility* is calculated in the following manner:

Mass of machine in grams = $1000 \times 454 = 4.54 \times 10^{5}$.
Area of base in square centimeters = $20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^{4}$.

Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^{4}} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of the entire support is $0.15 \times 10^{5} \times 1.86 \times 10^{4} = 0.28 \times 10^{9}$ mechanical ohms (or absolute units). Therefore,

$$\tau' = \sqrt{\frac{(0.28 \times 10^{9})^2 + \dfrac{1}{4\pi^2 \times 100^2 \times (0.134 \times 10^{-10})^2}}{(0.28 \times 10^{9})^2 + \left((2\pi \times 100 \times 4.54 \times 10^{5}) - \dfrac{1}{2\pi \times 100 \times (0.134 \times 10^{-10})}\right)^2}}$$

$$= \sqrt{\frac{0.0784 \times 10^{18} + \dfrac{10^{18}}{4\pi^2 \times 10^2 \times 0.018}}{0.0784 \times 10^{18} + \left(2\pi \times 4.54 \times 10^{7} - \dfrac{10^{8}}{2\pi \times 0.134}\right)^2}} = 0.935$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second.

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm. The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^{5} \times 645 = 0.97 \times 10^{7}$ mechanical ohms (or absolute units). Therefore

$$\tau'' = \sqrt{\frac{(0.97 \times 10^{7})^2 + \dfrac{1}{4\pi^2 \times 100^2 \times (0.39 \times 10^{-9})^2}}{(0.97 \times 10^{7})^2 + \left((2\pi \times 100 \times 4.54 \times 10^{5}) - \dfrac{1}{2\pi \times 100 \times (0.39 \times 10^{-9})}\right)^2}}$$

$$= \sqrt{\frac{0.94 \times 10^{14} + \dfrac{10^{14}}{4\pi^2 \times 0.1521}}{0.94 \times 10^{14} + \left(2\pi \times 4.54 \times 10^{7} - \dfrac{10^{7}}{2\pi \times 0.39}\right)^2}} = 0.0375$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the *transmissibility* is reduced to 0.0375, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

Controlling Noise Through Room Wall Surfaces

The ventilating equipment is usually housed in a separate room where the noise produced by the mechanical operation of the equipment can be isolated from the rest of the building. If the vibration of the machinery is absorbed by flexible mounting and is not transmitted to the building,

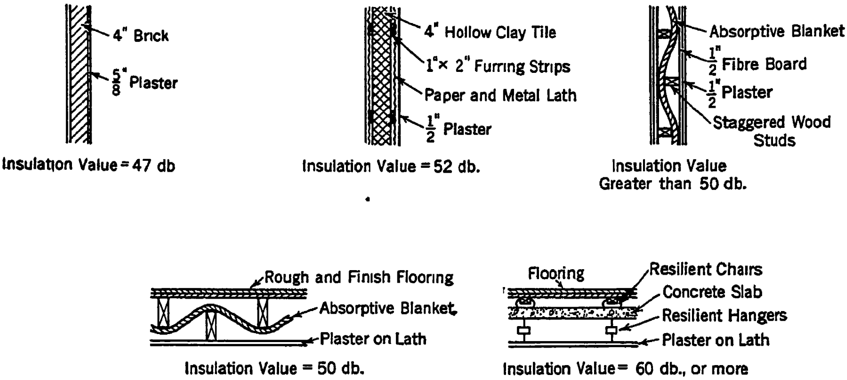

FIG. 1. THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH ARE SUITABLE FOR THE INSULATION OF EQUIPMENT ROOMS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1932, p. 211).

the only noise to be eliminated by the walls of the room will be the air-borne mechanical noise. Acoustical measurements on average brick, tile, lath, and plaster walls indicate that the usual wall of these types is sufficient to satisfactorily attenuate this air-borne mechanical noise.

Three wall sections and two floor and ceiling sections which are satisfactory for the wall insulation of the equipment room are shown in Fig. 1.

Attention should be given to the equipment room door, since this door may leak badly and allow sound to escape into parts of the building which should be quiet. Where the equipment noise is particularly severe, double doors should be used and in all cases, the doors of the equipment room should be fitted with tight thresholds and weather-stripping. The door itself may transmit considerable sound if it is thin but it will not transmit a tenth as much as will be transmitted by a ¼-in. crack between the door and the threshold.

In cases where the equipment noise is extraordinarily high, it may be

necessary to treat acoustically the walls and ceiling of the equipment room. If the equipment room is not entirely closed, partition walls may be necessary.

NOISE TRANSMITTED THROUGH THE DUCTS

After noise reaches the air stream in the ducts it can be controlled by lining the ducts on the inside with a sufficient quantity of sound absorbing material. Lagging material of similar characteristics placed on the outside of ducts serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Inside lining material used in the case previously mentioned would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream.

Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

To use the lining effectively it must be properly located, well installed and be applied in sufficient quantity to reduce the noise level of the air stream to the level desired.

At present there are no wholly rational or generally recognized methods of calculating the amount of duct lining necessary to accomplish a given reduction of noise level in the air travelling in a duct system; consequently some empirical method has to be used. One empirical method is to use direct trial and error. Another empirical method uses a duct lining factor evaluated by experience. In the present state of the data on sound control for ducts, the latter method is convenient for making estimates, but attention is specifically called to its empirical nature and to the necessity of exercising judgment in applying it.

Use of Duct Lining Factor

A duct lining factor (f) giving numerical values for use at various equipment noise levels is shown in Fig. 2. When properly used with Table 1 this chart (Fig. 2) provides a solution which may be both useful and simple. It is important to understand that the levels referred to in this chart are the *average* noise levels set up in the room by the ventilating or air conditioning equipment. In the case of a piece of equipment which generates a noise level of 95 db, when the noise is measured immediately

next to the machine, there might be a reduction of 15 db in passing through the duct, and a further difference of 15 db between the noise at the outlet supply grille and the average level in the room, leaving an effective level of 65 db in the room. Reductions of noise level ranging from 5 to 25 db through duct systems have been encountered without the use of sound absorbing linings and the drop from supply opening to average room level may vary from 5 to 20 db.

Duct lining factor f	Equipment		
	Noisy	Average	Quiet
0	75	65	55
5	65	55	45
10	55	45	35
15	45	35	25
20	35	25	15
25	25	15	5
30	15	5	-5

FIG. 2. CHART FOR DETERMINING NOISE REDUCTION IN DECIBEL[illegible] DUCT LINING FACTOR[a]

[a]Values for equipment noise are only general. Wherever possible substitute actual values as supplied by equipment manufacturer or as measured.

To determine whether to use column 1, 2, or 3 in Fig. 2, in forming an estimate of the relative amount of noise generated by the system, the length of the untreated duct system and the number of bends or elbows or splitters should be considered, since the longer and the more complex the system, the more reduction of noise level will occur before the sound reaches the room grilles. Also the sound absorbing power of the room should be taken into account, since in rooms where there is a great deal of absorptive material, such as rugs, draperies, curtains and furniture, there will be a higher loss between the outlet grille noise and the average room level. The ventilating engineer will have to judge whether the conditions deviate from the typical.

Manufacturers ratings on equipment should be considered in connection with the foregoing discussion. The quantity determined involves

the noise level which will be produced in the room and the manufacturer's method of rating must be considered before allowances previously mentioned are accepted.

To use Fig. 2, proceed by consulting Table 1 and determine the probable noise level already existing in the room, and, as suggested, assume that this level is satisfactory for current practice. This gives a noise level in decibels and with this enter the chart of Fig. 2. Read across the chart and determine the value of the duct lining factor (f) in the column at the left. Then multiply the smallest cross sectional dimension (inches) of the duct by this factor. The result will be the length of duct in inches to be lined to attenuate an average fan noise. If circular ducts are used, the length to be lined will be (f) $\times$ diameter of duct.

Example 2. A 7 x 30 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate a fan noise satisfactorily.

From Table 1 the noise level in this office will be 35 db.

Length to be lined for noisy equipment is $22 \times 7 = 154$ in.
Length to be lined for average equipment is $17 \times 7 = 119$ in.
Length to be lined for quiet equipment is $12 \times 7 = 84$ in.

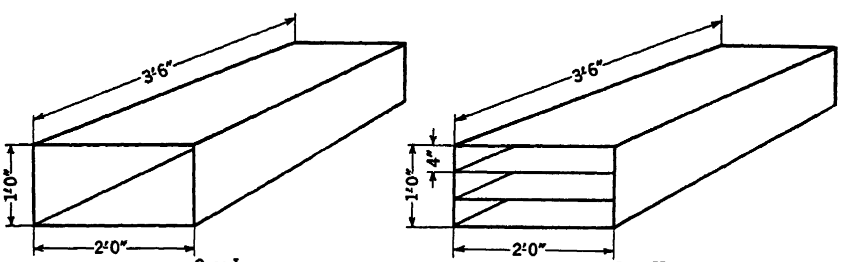

FIG. 3. DIAGRAM OF BRANCH DUCT CONNECTION

The sound absorbent properties of duct lining are extremely important and materials which have coefficients as high as possible should be used. This is particularly true of the coefficients at the low frequencies. Fig. 2 is based on materials having a noise quieting coefficient of 0.60 or more. For materials which are less efficient a factor of safety should be added[4].

Only certain sound absorbent materials among those listed in various publications will be found to be suitable for duct lining. In addition to a high sound absorbent coefficient a duct lining material should have a low surface coefficient of friction, high resistance to moisture absorption, and should be fireproof and vermin proof. A number of building codes now specify that any sound absorbent material used for duct lining shall have no fire hazard. There are no existing specifications on moisture resistance but the manufacturer should be required to show that the material will not absorb sufficient moisture to cause deterioration or to decrease the sound absorbing efficiency.

[4]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Manufacturers' Association,* 919 No. Michigan Ave., Chicago, Ill.

If, as is often the case, the length of duct from the main duct to a grille is shorter than the length of lining indicated by using the factor found, this duct may be sub-divided[5] into smaller ducts, so that the value found may be used as shown in Fig. 3.

Example 3. Assume a branch duct, as shown in Fig. 3, is 24 in. wide by 12 in. high and 42 in. long. Use a duct lining factor of 10.

Case I. (No splitters).

Length of lining $= f \times$ minimum dimension $= 10 \times 12 = 120$ in.
In this case the duct should be lined for 120 in. which is obviously impossible.

Case II. (Two splitters).

Results in 3 ducts 24 in. wide and 4 in. high.
Length of lining $= f \times$ minimum dimension $= 10 \times 4 = 40$ in.
This length of lining fulfills the space limitations of the branch duct which is 42 in. long.

General Suggestions

In some instances where high velocity air is used, a considerable amount of whistle is generated at the grille. This noise is obviously produced after the air leaves the duct and there is no treatment which can be installed in the duct that will reduce this noise. The engineer must take into consideration the type of grille which he intends to use and provide sufficient grille area so that the velocity through the grille is reduced to a point where the grille is not too noisy.

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment.

Very often in ventilating duct work the engineer feels that it will not be necessary to line ducts if the sound is travelling against the airflow. This, however, is untrue since sound travels so much more rapidly than does the air in even high velocity systems, that it will travel as easily against the airflow as it does with it.

Sounds which are low in pitch are much harder to eliminate from a duct system than sound which is high in pitch, consequently equipment which produces low pitched sounds should be avoided as much as possible.

REFERENCES

How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning* October 1931, p. 815).

The Nature of Noise in Ventilating Systems and Methods for Its Elimination, by J. S. Parkinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning* March, 1937, p. 183).

Effect of Humidity upon the Absorption of Sound in a Room, by V. O. Knudsen (*Journal, Acoustical Society of America,* July 1931). Also see report presented at the May 1933, meeting of *A.S. of A.*

Acoustics and Architecture, by P. E. Sabine.

Architectural Acoustics, by V. O. Knudsen.

Acoustical Engineering, by West.

Modern Acoustics, by Davis.

[5]Patents exist covering the sub-dividing of ducts for installing sound absorbent materials.

PROBLEMS IN PRACTICE

1 ● Does a soft pad under the ventilating machinery prevent building vibration?

It may or it may not. In some cases a soft pad causes more vibration than no pad. A flexible mounting should be carefully designed to be effective.

2 ● Are especially designed walls necessary in an equipment room to keep noise out of adjoining spaces?

Ordinarily good brick, tile, or concrete walls are satisfactory. Window and door openings should be made as tight as possible with weather-stripping, etc.

3 ● How should mechanical noise be eliminated from the duct system?

A flexible connection between the fan discharge and the duct should be used. The duct should be lined from the fan end for a certain length, depending on the degree of quietness desired.

4 ● Given the choice of two types of equipment, one generating high-pitched sounds and the other low-pitched sounds, which would you choose? Why?

The equipment generating the high-pitched noise should be chosen since high-pitched sounds are more easily absorbed than are low-pitched sounds.

5 ● In building an acoustic filter in a short duct 32 by 24 in. which direction should the splitters run?

The splitters should be installed parallel to the longest dimension, since they will provide more acoustical material per splitter.

6 ● What should be the characteristics of a good duct-lining material?

a. High noise reduction.
b. Physical strength.
c. Easy working and installation.
d. Fire resistance.
e. Cleanliness, absence of loose fibers or pieces.
f. Smooth surface to reduce air friction.

7 ● Should a ventilating duct be lagged or covered on the outside?

Yes, in some locations, and particularly in the equipment room and where the duct runs through noisy rooms to serve a quiet room. This lagging will prevent air-borne sounds from entering the duct through its sides and causing annoying sound in the quiet room.

8 ● How can cross-talk be eliminated when one duct serves two or more rooms?

Install proper filters adjacent to the grilles in each room, using splitters if the duct leads to the rooms are short.

9 ● Space limitations and maximum air velocities for the introduction of air to a broadcasting studio restrict the size of duct to 30 by 16 in. and in addition the length of branch duct which is suitable for lining with sound absorption material is limited to 22 ft. Determine the length of duct lining necessary to attenuate an average fan noise and establish a permissable room noise level.

Referring to Fig. 2 the noise level for broadcasting studio is 14 db and the corresponding duct lining factor f is 28. Minimum cross sectional dimension of duct = 16 in.

$$\frac{16 \times 28}{12} = 37.3 \text{ ft duct lining required.}$$

Maximum length of duct is 22 ft, therefore it is necessary to divide the duct with a splitter, resulting in a minimum duct dimension = 8 in.

$$\frac{8 \times 28}{12} = 18.7 \text{ ft duct lining required to attenuate an average fan noise.}$$

Chapter 31

AIR CONDITIONING IN THE TREATMENT OF DISEASE

Operating Rooms, Reducing Explosion Hazards, Post-operative Heat Stroke, Nurseries for Premature Infants, Fever Therapy, Control of Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

IN the past few years air conditioning has made considerable progress as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, in heat therapy, oxygen therapy, X-ray rooms, and in the control of allergic disorders.

AIR CONDITIONING OPERATING ROOMS

The most wide application of air conditioning in hospitals is that in operating rooms. Complete air conditioning of operating wards is not only desirable but often necessary for reducing the risk of explosion of modern anesthetic gases in dry winter atmospheres, and for the protection of the patient and operating personnel against excessive summer heat.

Reducing Explosion Hazard

Explosion hazards in operating rooms have begun with the introduction of modern anesthetic gases and anesthesia apparatus. Ether administered by the old drop method is still regarded as comparatively safe; but when mixed with pure oxygen or with nitrous oxide in certain concentrations (see Table 1) the explosion hazard may be as great as with ethylene-oxygen mixtures.

During the course of ethylene anesthesia the mixture, usually 80 per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight, confined to an area in the immediate vicinity of the face mask, where leakage of ethylene into the air may accumulate to the lower explosion concentration (see Table 1). The most dangerous period is at the end of the operation when the patients' lungs and apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration of anesthesia. In either case the mixture would pass through the explosion range and extra-

ordinary precaution is necessary for the safety of the patient and operating personnel.

Copious ventilation, from 6 to 12 air changes per hour, is necessary to preclude accumulation of explosive mixtures and to reduce the concentration of anesthetics to below the physiologic threshold so that the surgeon and his personnel will not be affected.

The most important cause of accidents is probably static sparks which may result from accumulation of frictional charges on the rubber surfaces of the anesthesia apparatus, on woolen blankets, and on the bodies of the operators as they walk on insulated floors, when the humidity is quite low. Grounding the various parts of the anesthesia apparatus is not entirely effective, so long as rubber remains in use in the conventional equipment.

To prevent accumulation of static charges within the apparatus or on persons coming near to it, the measures proposed[1] are humidification of air to between 55 and 60 per cent relative humidity, grounding the

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF ETHYLENE AND ETHER[a]

Mixed With	Ethylene		Ether	
	Lower Limit Per Cent	Upper Limit Per Cent	Lower Limit Per Cent	Upper Limit Per Cent
Air	3.0	30 ±	1.7	50 −
Oxygen	3.0	80 −	1.7	40 ±
Nitrous Oxide	-----	-------	3.8	26 ±

[a]Limits of Inflammability of Gases and Vapors, H. F. Coward and G. W. Jones, *U. S. Department of Commerce, Bulletin No.* 279, 1931.

apparatus and operating table, and using conducting floors and shoes so that the operating staff and attendants will be always grounded as they move about. The significant factor is the absolute humidity, rather than the relative humidity, because upon it depends the electrical conductivity of the atmosphere. The principal objection to artificial humidification is the necessity of constant supervision to make sure that the apparatus is functioning properly.

Artificial humidification in operating rooms during cold weather may also prove beneficial in reducing evaporation from exposed tissues and from the wet skin of the patient, and by allowing a lower room temperature.

Operating Room Conditions

Little is known about optimum air conditions that are necessary to maintain a normal body temperature during the course of anesthesia and in the immediate post-operative period.

Under the influence of anesthesia a patient is at a very low ebb. All anesthetics, as a rule, produce dilation of the vessels in the skin and much sweating, particularly in the case of ether anesthesia. The loss of body heat is increased considerably, while the general metabolism may be

[1]The Hazard of Explosion of Anesthetics, by Y. Henderson. Report of the Committee on Anesthesia (*Journal American Medical Association*, 94:1491, 1930).

depressed. The organism loses ability to regulate its own body temperature and becomes unusually sensitive to chilling and post-operative complications. In order to maintain a normal body temperature, a high air temperature is necessary, as high as 90 F or higher in the case of ether anesthesia, judging from experiments on animals[2].

Such high temperatures are obviously uncomfortable for the operating personnel, and in order to alleviate the condition the room temperature is usually kept between 72 and 80 F in cold weather with the patient carefully guarded with blankets and hot water bottles during and for some time after the operation.

Post-operative Heat Stroke: It would seem that surgeons have learned to fear so much the occurrence of post-operative pneumonia and shock that even in hot summer weather patients are sometimes needlessly *bundled up* with detrimental consequences.

In 1916 several deaths were reported[3] of heat stroke following surgical operations, and a number of cases suffering from a mild isolation, often recognized as post-operative reaction or shock. From these observations it was concluded that all operating room activities should cease during summer heat waves with the exception of urgent operations, when every effort should be made to keep the patient cool and comfortable.

In cases of exophthalmic goitre, one investigator[4] warns most emphatically against the performance of operations in extremely warm weather, for under such conditions the risk in spite of all precautions (prior to the introduction of summer cooling in operating rooms) is too great. An analysis of several cases over a 10-year period shows a striking rise of post-operative deaths in June, July, and August, resulting unexpectedly from extreme post-operative reaction passing onto acute hyperthyroidism.

More recently four cases were reported[5] of post-operative heat stroke admitted 24 hours preceding operation and sheltered from direct sun rays. All four were not ill and apparently were good risks. There occurred, however, at the time of operation and for several days preceding it, a heat wave with a moderately high temperature, a high relative humidity, and no wind. In addition to warm weather, excessive loss of body fluids is believed to have been a factor in the production of heat stroke in those four cases.

Aside from the possibility of post-operative heat stroke in warm and sultry weather, the surgeon is also concerned with the lowered recuperative power of the patients, and with his own discomfort as well as the discomfort of his team, which impairs the efficiency of the technic to the disadvantage of the patient.

In view of this experience it is customary to defer major operations as much as possible until the passing of heat waves, in hospitals not equipped with cooling facilities. But there are exceptional cases, like acute appen-

[2]Heat Regulation and Water Exchange. The Influence of Ether in Dogs, by H. G. Barbour and W. Bourne (*American Journal Physiology*, 67:399, 1924).

[3]Post-operative Heat Stroke, by A. V. Moschcowitz (*Surgery, Gynecology and Obstetrics*, 23:443, 1916).

[4]The Effect of Heat Upon Operations for Exophthalmic Goitre, by A. J. Walton (*British Medical Journal*, 1:1045, 1923).

[5]Post-operative Heat Stroke, by T. M. Martin (*Journal Missouri Medical Association*, July, 1928. *Abstract Anesthesia and Analgesia*, 8:23, 1929).

dicitis for instance, which sometimes come with summer heat waves, and develop dangerously unless promptly operated upon. Complete air conditioning of operating rooms would therefore seem to be a necessity in many sections of the United States.

Satisfactory Air Conditions: Although the comfortable air conditions for the operatives are not identical with those of the patient, a compromise is as a rule not difficult; with a relative humidity of 55 to 60 per cent, a temperature of 80 F in warm weather and between 72 and 75 F in cold weather will probably prove satisfactory. Additional heat may be furnished to the patient locally or by suitable covering according to body temperature in individual cases.

Central station air conditioning plants and individual unit air conditioners proved satisfactory in operating rooms when producing between 8 and 15 air changes per hour of filtered and properly humidified air, with full provision for summer cooling and dehumidification and without recirculation during the course of anesthesia. A separate exhaust fan system is as a rule necessary in order to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather and to minimize drafts. The high air flow of 8 to 15 air changes in operating rooms is desirable for three reasons: (*a*) to reduce the concentration of the anesthetic to well below the physiologic threshold in the vicinity of the operating personnel, (*b*) to remove excessive amounts of heat and sometimes moisture from sterilizing equipment if inside the operating room, from the powerful surgical lights, solar heat, and from the bodies of the operatives, and (*c*) to provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by careful insulation of sterilizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

It is generally believed that in addition to operating rooms, an adjoining ward should also be conditioned to provide for the treatment of post-operative fever. Such a post-operative ward may also prove valuable in treating patients with heat stroke, fevers, summer diarrhea and other cases affected by high temperature, when the room is not used for anything else.

Sterilization of Air in Operating Rooms: Of considerable significance to operating rooms and contagious wards is the use of ultra-violet radiation for sterlizing the air.[6] Results reported[7] would seem to indicate that the post-operative temperature rise of patients during the first few days is in most instances caused more by bacterial contamination of the operative wound than by the absorption of blood and traumatized tissues. Operating room infections, which were quite frequent before the installation of special ultra-violet lamps, are said to have practically disappeared.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because

[6]Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, 17:253, 1925).

[7]Sterilization of the Air in the Operating Room by Special Bactericidal Radiant Energy, by Deryl Hart (*Journal Thoracic Surgery*, 6:45, 1936).

their heat regulating system is not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain a normal body temperature by their own efforts. The resistance to infection is low and the mortality rate, very high.

Air Conditioning Requirements

The optimum air conditions for the growth and development of these infants were determined by extensive research at the Infants Hospital, Boston, Mass.,[8] using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Wide variations were found in individual requirements for temperatures from 72 to 100 F, according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F temperature and 65 per cent relative humidity was found to satisfactorily fulfill the requirements of the majority of premature infants. Additional heat for weak or debilitated infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Infants Hospital were kept at relative humidity between 25 and 50 per cent for two weeks or longer, the body temperature became unstable, gains in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years.

The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery, minimum in the conditioned nurseries under high humidity, and intermediate in the conditioned nurseries with low humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea,

[8]The Premature Infant: A Study of the Effects of Atmospheric Conditions on Growth and on Development, by K. D. Blackfan, C. P. Yaglou and K. McKenzie (*American Journal Disease of Children*, 46:1175. 1933).

persistent vomiting, diminishing gains or loss of body weight, and other symptoms, were generally from two to three times as high under low than under high humidity.

Finally, the mortality of premature infants was found to be greatly affected by humidity. In Table 2 is given the net mortality according to the humidity in which the derangement of body function began. In the old nurseries, prior to the installation of the air conditioning system, the death rate from acute and chronic infections was 26.5 per cent as compared with 9.7 per cent in the conditioned nurseries under low humidity and 0.0 per cent under high humidity.

Summarizing the conclusions of these studies, the best chances for life in premature infants are created by maintaining a relative humidity of

TABLE 2. NET MORTALITY OF PREMATURE INFANTS ACCORDING TO HUMIDITY[a]

Infants Hospital, Boston, Mass.

Cause of Death	Unconditioned Nurseries (1923-1925)	Conditioned Nurseries (1926-1929)	
	Natural Humidity	Relative Humidity	
		25-49 Per Cent	50-75 Per Cent
	Per Cent Mortality	Per Cent Mortality	Per Cent Mortality
Acute and chronic infections	26.5	9.7	0.0
Congenital deformities	1.2	0.0	0.7
Unclassified	1.2	4.8	0.0
All causes	28.9	14.5	0.7

[a]Excluding cases with multiple congenital anomalies incompatible with life, and also deaths occurring within 48 hours after admission to the hospital.

65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment: Most of the installations now in use are of the central station type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A high ventilation rate, between 15 and 25 air changes, is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation is not used extensively in these wards owing to odors and the possibility of infection.

AIR CONDITIONING IN FEVER THERAPY

Artificial production of fever in man is an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by obliteration or destruction of the invading organism within the safe limit of human fever temperature; or an indirect one in case of heat resistant organisms, through general mobilization of the defensive

mechanisms of the body, by means of which the activity of pathogenic bacteria and their toxins may be retarded or neutralized.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 6 hours at a time. The total period of fever treatment varies with the type of the organism involved from a few hours to 50 hours or more.

The diseases reported to respond favorably to artificial fever are: gonorrhea, neurosyphilis, chorea, asthma, peripheral vascular diseases, occular gonorrhea and syphilis. There are a number of other conditions in which the usefulness of artificial fever is not yet settled. The most striking results are seen in gonorrhea in which various strains of organism can be killed by artificial fever within the limits of tolerance of man.

Equipment for the Production of Systemic Fever

Various means have been tried for producing artificial fever, including injections of various crystalloid or colloid substances; a number of physical methods, such as hot baths, radiant heat, diathermy, radiothermy, and, in the last few years, an air conditioned chamber. The relative advantages and disadvantages of these various methods were discussed in recent papers.[9] The results by the use of air conditioned cabinets have not been fully explored, and it is therefore difficult to determine the advantages and disadvantages of the value of air conditioning at this time. Under certain conditions a combination of systemic fever and additional local heating by diathermy or other means is claimed to yield better results than systemic fever alone by reducing considerably the killing time of the organism and rendering the treatment less trying to both patient and attendants.[10]

The air conditioned chamber[11] consists of an insulated cabinet approximately 6 ft long, 3 ft wide, and 2.5 ft high, containing in a small rear compartment, electric air heaters, a water pan for humidification, a centrifugal fan, and controls. The nude patient lies on an air mattress inside the front compartment with his head protruding outside the front end through a rubber collar. Warmed air at 130 to 150 F and 30 to 50 per cent relative humidity is blown upon the body of the patient, and the rectal temperature rises to 105 F usually in from 40 to 60 min. The heat is then turned low and adjusted so as to maintain the desired body temperature in each individual case.

More recently a heat cabinet was described[12] in which saturated air between 100 and 120 F in temperature is used for elevating the patient's body temperature. This gives a rapid rise of body temperature with a relatively low air temperature; it eliminates skin burns, and the room in which the heat box is located is not overheated unduly.

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[9]Fever Therapy for Gonococcic Infections, by A. U. Desjardins, L. G. Stuhler and W. C. Popp (*Journal American Medical Association*, 106:690, 1936).
Artificial Fever Therapy as a Therapeutic Agent, by H. P. Doub (*Radiology*, 25:360, 1935).

[10]The Treatment of Gonorrheal Arthritis by Means of Systemic and Additional Focal Heating, by W. Bierman and C. Levenson (*American Journal Medical Science*, 191:55, 1936).

[11]Artificial Fever Therapy of Syphilis, by W. M. Simpson (*Journal American Medical Association*, 105:2132, 1935).

[12]Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1937, p. 115).

Extensive research is now in progress to determine the usefulness and limitations of fever therapy on a wide variety of pathogenic conditions. While this form of therapy is rapidly gaining wide recognition, its application, according to the *American Medical Association*, should be strictly a hospital procedure surrounded with the safeguards commonly employed in a major operation and under the direction of skilled physicians.

CONTROL OF ALLERGIC DISORDERS

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to foreign or pollen proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption of the allergens through the skin. The clinical manifestations are hay fever, asthma, eczema, hives, contact dermatitis, etc.

Symptoms of Hay Fever and Asthma

The respiratory tract is probably the most usual site of allergic manifestations, the so-called hay fevers and asthma. In hay fevers, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and most distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors and irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals is likely to be unstable, with a tendency to subnormal body temperature. Many allergic cases who are apparently well, develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who failed to respond to medical treatment (desensitization or immunization).

Paper or cloth filters, mounted in inexpensive window or floor units, proved quite satisfactory in removing all but traces of pollen. Allergens may also be removed by passing the air through a water spray, or over cooling coils kept at a temperature low enough to cause condensation of atmospheric moisture on the surface of the coils.

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification

appears to be desirable. A comfortable temperature between 75 and 82 F in warm weather and a relative humidity well below 50 per cent proved satisfactory.[13] Direct drafts, overcooling or overheating are apt to initiate or aggrevate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. But while specific desensitization is preventive and in a few instances curative for all practical purposes, filtration gives only temporary relief. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond at all to the treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief. Hay fever cases are usually relieved of most of their symptoms within an hour or so after exposure to properly filtered air. In pollen asthma cases relief comes more slowly, usually after an exposure of from 1 to 12 days depending upon the severity of asthma.

A pollen free atmosphere is especially valuable for patients in whom desensitization has given little or no relief, and in instances in which desensitization is not advisable owing to intercurrent illness. On the whole, conditioning methods are considered to be a valuable adjunct in medical diagnosis and treatment of allergic disorders.

AIR CONDITIONING IN OXYGEN THERAPY

Oxygen therapy is the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants.

The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. The oxygen rich atmosphere in these enclosures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants.

Oxygen Tents: In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately

[13]The Effect of Low Relative Humidity at Constant Temperature on Pollen Asthma, by B. Z. Rappaport, T. Nelson and W. H. Welker (*Journal Allergy*, 6:111, 1935).

cooled oxygen tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good[14]. An ice melting rate of about 10 lb per hour gives satisfactory results in patients with fever in a medium size tent.

Oxygen tents are somewhat confining to the patient; the restless type of person is difficult to control, and the delirious impossible to control. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. The direct advantages are portability and low cost.

Oxygen Chambers: The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from electric fixtures outside the chamber.

The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or silica gel for drying the air. The gravity system includes a bank of brine coils controlled thermostatically, which dehumidify and cool the air. The cool air falls over trays at the bottom of the coils, containing soda lime to remove the carbon dioxide given off by the occupants. A heater at the base of the opposite wall warms the air to the desired temperature. Ordinary industrial oxygen is introduced from storage tanks outside the chamber and the concentration is regulated according to the prescription of the physician. The only change of air in the chamber is that taking place by leakage through trap doors.

The chief objections to the gravity circulation system are stratification of cold air near the floor and accumulation of odors, which may require the use of activated charcoal or an excess of oxygen for deliberate aeriation.

The fan circulation systems include compact extended surface coolers, heaters, and sometimes silica gel beds installed outside the chamber for the removal of moisture. A spray dehumidifier is not suitable for this purpose because it is often desired to cool the air below 32 F in order to obtain low relative humidities.

The temperature and humidity requirements in oxygen therapy depend primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias, the range of satisfactory conditions is placed between 60 and 75 F with 20 to 50 per cent relative humidity, depending on the condition of the patient.

Oxygen chambers are unquestionably more comfortable than oxygen tents. The patients receive unhampered medical and nursing care, and the oxygen concentration, the temperature and humidity can be adequately controlled at any desired level. The chief disadvantage is high initial and operating costs in comparison with oxygen tents or with the nasal catheter method of oxygen administration. The nasal catheter

[14]General Measures Employed in the Treatment of the Pneumonias, by J. G. M. Bullowa (*Health Examiner* 5:12, 1936).

method is the simplest and most inexpensive of all but it may cause considerable discomfort to the patient and it is not satisfactory for continuous administration and in restless or delirious patients. Moreover, oxygen concentrations greater than 40 per cent in the inspired air are difficult to obtain.

The chamber method is of value in large hospitals and for research and experimental purposes, but for routine oxygen therapy alone it may prove a liability rather than an asset in many hospitals.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of large hospitals involves a capital investment, depreciation and running expense which may not be justified.

In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of rational heating in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help considerably in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the window openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire hospital, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North and certain sections of the Pacific Coast, cooling is needed on but a few days during summer, while in the South, built-in room coolers can be used to advantage from May to October, and in tropical climates almost continuously throughout the year. Objectionable noise is an important drawback to the use of self-contained units, but the difficulty is gradually being overcome by improvements in design.

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of pyrexias in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is now in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyperthyroidism, essential hypertension, skin diseases, vascular disorders, and others. The field is a fruitful one having many possibilities.

PROBLEMS IN PRACTICE

1 ● Where has air conditioning in hospital wards proved itself of sufficient value to justify the expense?

In nurseries for premature infants, anesthesia and operating rooms, oxygen therapy chambers, heat therapy rooms or cabinets and allergic wards.

2 ● What is the major problem in conditioning hospitals?

For general hospital wards, the major problem seems to be one of providing adequate amounts of ventilation rather than air conditioning, with some provision for cooling over-heated wards on unusually warm summer days.

3 ● What are the usual requirements for ventilation of operating rooms?

To preclude the accumulation of explosive mixtures and to reduce the concentration of anesthetics below the physiologic threshold, it is desirable that ventilation to the extent of 6 to 12 air changes be provided.

4 ● What are the optimum air conditions for premature infants?

The best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent and an environmental temperature sufficiently high to keep the body temperature within normal limits.

5 ● How does air conditioning assist in the treatment of allergic disorders?

In cases in which the individual fails to respond to medical treatment, air conditioning may provide a valuable adjunct in relieving the symptoms. Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, it has been found that in warm weather temperatures between 75 and 82 F and relative humidities well below 50 per cent are more conducive to comfort.

Chapter 32

RAILWAY AIR CONDITIONING

Passenger Car Ventilation, Quantity of Outside Air, Method of Air Distribution, Air Cleaning, Steam or Vapor Heating Equipment, Cooling Equipment, Humidity and Temperature Control, Power Supply, Installation and Operating Costs

THE general principles of air conditioning as applied to buildings also apply to railway passenger cars, but due to space and weight limitations and the severity of the service, equipment designed for stationary work is seldom suitable for car installations. Equipment foi railway use must be safe, reliable, compact, light in weight, accessible for inspection and repairs, automatic in operation and in addition, have low initial, operating, and maintenance costs. To air condition a passenger car properly, ventilating, filtering, heating, cooling, humidifying, and control equipment must be provided together with an adequate power supply. Air from the interior of the car is mixed with air from the outside and passed through the air conditioning unit where it is heated or cooled, humidified or dehumidified and delivered to the interior of the car through suitable ducts and grilles.

PASSENGER CAR VENTILATION

One of the important problems in connection with air conditioning of cars is that of ventilation. In non air-conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators and open doors and windows. This practice provides an ample supply of outside air but does not prevent the entrance of smoke, cinders, and dirt.

Quantity of Outside Air

An average passenger car contains approximately 5000 cu ft of air and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but the other constituents can be successfully handled only by proper ventilation and air cleansing. In the average car from 2000 to 2500 cfm should be circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from the outside. The amount of outside air required depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of the total air circulated. The per-

centage of outside air should be kept as low as possible to maintain the air in the proper condition in order to minimize the heat or cooling load.

For normal conditions, 10 cfm of outside air per passenger is sufficient. When smoking is permitted, 12 to 15 cfm per passenger should be admitted. Under exacting demands and adverse condition, it may be necessary to increase the quantity to 20 cfm per passenger.

Method of Air Distribution

Various methods may be used to distribute the air delivered to the interior of the car by the circulating fan or blower. The methods commonly used are:

1. A duct lengthwise along the center of the car.
2. One or two side ducts built on the outside of monitor-roofed cars, or on the inside of turtle-backed or arched-roofed cars.
3. Free discharge at the end bulkheads, or by free discharge from a unit placed overhead in the center of the car, discharging toward the ends.

For details of air distribution and duct design, see Chapters 28 and 29.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the washroom or lavatory, and then outside by a fan in a ventilator.

For compartments an adjustable supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for the removal of the air to be recirculated.

The grille used for this purpose should be designed and arranged so as to obstruct the vision of passengers, but still allow the air to pass from the room to the recirculating grille at the air conditioning unit.

Lower berths in sleeping cars and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Care must be exercised to secure proper circulation, otherwise objectionable drafts will be experienced.

The recirculating air grilles are usually of the straight flow type, and should be located so that objectionable drafts will not be created by the return air. The outside air intakes, located in the car vestibule, on the side of the car, or on the roof of the car, depending upon the location of the cooling coils, should be of ample size to permit the entrance of sufficient outside air. On many of the recently air-conditioned cars, there are no dampers or shutters at the outside air intakes, the percentage of outside air being controlled by blocking the flow through the recirculating grille.

Air Cleaning

All of the air circulated by the blower is filtered before passing over the cooling coils. In some cars the outside and recirculated air are filtered separately before mixing, while on others the air from the two sources is mixed before passing through a common filter. Filters in use are made of metal, wool, cloth, spun glass, hemp, paper, hair, and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, retreated, and returned to service while other types are discarded when dirty.

STEAM OR VAPOR HEATING EQUIPMENT

The majority of cars in service are heated by circulating low pressure steam or vapor through pipes located along the side walls near the floor. When an air conditioning unit, using air from the outside, is installed it is necessary to provide a heating coil to warm the air during cold weather. Usually from 30 to 40 per cent of the heat required is supplied from the air conditioning unit and the balance from the floor heating system. It is necessary to have sufficient floor radiation to keep water lines in the car from freezing while standing in the yard with the air conditioning unit shut off. In new and some rebuilt cars, finned pipes are used for the floor heat to provide greater radiation surface. A few cars have the heating pipes enclosed in a duct, through which air is forced by a fan, the warmed air discharging through numerous openings along the floor. The amount of heat required depends upon the type and construction of the car, especially the amount and kind of insulation, outside temperature, wind velocity and direction, train speed, number of passengers, and inside temperature desired. In severe weather, with temperatures from −10 to −20 F, an average of approximately 200 lb of steam per car per hour is required. Pullmans require approximately 250 lb per hour, coaches 150 to 175 lb per hour and baggage cars 150 lb per hour.

COOLING EQUIPMENT

Three general types of cooling or refrigerating equipment are being used with satisfactory results. These are the ice-activated, the steam-ejector and the mechanical compression systems. These systems when arranged for car use, function the same as in stationary service, but must be more compact and lighter in weight. See Chapter 24 for description of the general principles of the various systems.

The mechanical compression systems are divided into three general classes depending upon the type of drive for the compressor, namely, the electro mechanical, the direct mechanical, and the internal combustion engine mechanical. The compressor of an electro mechanical compression system is driven by an electric motor, the power for which is supplied by a generator and a storage battery. The generator is driven from the car axle by a gear, belt, or other type of mechanical drive. The compressor of a direct mechanical compression system is driven directly from the car axle by means of a mechanical drive, the speed of the compressor being regulated by an electric speed control which permits slippage at high train speeds. The compressor of the third type of mechanical compression

system is driven by an internal combustion engine operating on propane. Sufficient fuel for several days' operation is carried in drums mounted in a rack under the car.

The refrigerant frequently used in the mechanical compression systems is dichlorodifluoromethane. The condensers are cooled by blowing a large quantity of outside air over the dry condenser coils, or over the coils wet by a spray of water to obtain the benefits of evaporative cooling. The latter method gives lower discharge pressures which is a distinct advantage when operating at high outside temperatures. A device gaining in popularity is a liquid subcooler by means of which the liquid refrigerant is subcooled by evaporative cooling, producing more available refrigeration at the air conditioning unit.

Another type of system which has been tried for railway passenger car air conditioning uses dry ice as the refrigerant, the equipment being essentially the same as for the water ice-activated system. Until an adequate supply of dry ice can be assured at a stable and reasonable price, this system will not be a serious competitor to the other three types now in use.

The capacity required in the refrigerating system depends upon a number of factors such as size and type of construction of car, thickness and kind of insulation used, the amount of heat produced within the car by motors, lights, and other appliances, the amount of outside air, the intensity of solar radiation, the number of occupants, and the inside temperature desired. The sun load on a bright day is about 1.2 tons. For average cars on sunny days, with high outside temperatures and humidities, from 65,000 to 80,000 Btu per hour will have to be removed from the interior of the car to maintain an inside effective temperature within the comfort zone. This means that a refrigerating capacity of from 5.5 to 7.0 tons will be required.

HUMIDITY CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the humidity desired inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely on the temperature of the evaporator, which should be below the dew-point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew-point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed. Such a system is costly and has not been generally applied.

During the heating season humidification is desirable from a comfort standpoint, but unless properly controlled, condensation will appear on

the windows. A steam or water spray controlled by a humidistat will provide the necessary moisture for humidification. There are several cars with this feature now in use.

TEMPERATURE CONTROL

The control of the air conditioning equipment should be simple and automatic in order to eliminate the human element for the selection of the control point. The use of a centralized panel for all switches, fuses, relay, etc., will simplify the installation and operation. Generally, separate thermostats are used for heating and cooling control. The best location for the thermostats depends upon the car layout and method of air distribution and can best be determined for any particular type of car and equipment by careful consideration of the several factors involved. The floor heat thermostats are usually located near the floor. The overhead heat and cooling thermostats are placed in the upper part of the car, sometimes in the air ducts or at the recirculating grille. All thermostats should be located so that the air can circulate freely around them. Maintenance of uniform comfort conditions for cooling, floor and overhead heating, has been satisfactory with provisions for a high, a medium, and a low thermostat setting and in some cases two settings have been satisfactory for cooling. In many cars the following points have been found to be satisfactory: 72, 74, and 76 F for cooling, and 60, 71 and 74 F for floor and overhead heating.

A few cars are in operation in which the inside temperature is varied dependent upon the outside temperature in order to prevent too high a differential between inside and outside temperatures. The maximum inside dry-bulb temperature permitted by these controls is usually 80F.

The heating and refrigerating equipment should be interlocked so that they cannot both operate at the same time. While heating, the control should be so arranged that in case of a steam failure the blower fan will stop or the outside air intake should be closed to prevent cold outside air from being introduced into the conditioned space.

POWER SUPPLY

One of the most important problems to be solved in connection with railway car air conditioning is that of power supply. The majority of non air-conditioned cars now in service are electrically lighted and equipped with fans. Power is furnished by storage batteries and axle generators of from 2 to 5 kw capacity.

Electric Power Requirements

When air conditioning is installed the electrical load is increased, according to the type of system as indicated in Table 1. To furnish this additional electric power, the capacity of the axle generators must be increased to 4 to 20 kw, and the storage battery capacity increased, in addition to that required by the car lighting system by 400 to 700 amp-hr for the electro-mechanical system, by 150 to 300 amp-hr for the steam-ejector and direct drive mechanical systems, and by 50 to 200 amp-hr for the ice-activated and internal combustion engine mechanical systems.

TABLE 1. ELECTRIC POWER REQUIRED TO OPERATE SYSTEM

System	Kilowatts
Electro-Mechanical	10.50
Direct Drive Mechanical	1.00
Internal Combustion Engine Mechanical	1.25
Steam-Ejector	3.35
Ice-Activated	1.20

Total Power Requirements

In addition to the electric power requirements, for continuous operation at average temperatures, the direct drive mechanical system requires 10.24 hp from the car axle and the steam-ejector system requires 230 lb steam per hour from the locomotive boiler for a 6-ton unit. The ice-activated system requires 463 lb of ice per hour and the internal combustion engine drive mechanical requires 7.3 lb propane per hour. This power, with the exception of the ice and propane, as well as the power required to move the extra weight of the equipment and the power required to overcome the axle bearing friction, must be supplied by the locomotive en route, and if a number of cars in the train are air conditioned, the effect on train performance should not be overlooked. The demand for power for cooling comes, however, at the time of the year when steam for heating is not required, and the demand for lighting is at a minimum.

The total power required by the air conditioning systems will vary with the speed of train operation because of the effect of speed upon the drive efficiency and upon the resistance due to the added weight of the equipment. Fig. 1 shows the effect of speed upon the efficiency of the direct drive used with the direct mechanical system, and upon the average efficiency of four mechanical drives and generators used for electric power generation. The total increase in weight of passenger cars because of air conditioning is approximately 9,600 lb for the electro-mechanical, 8,600 lb for the direct mechanical, 8,600 lb for the internal combustion engine drive mechanical, 11,300 lb for the steam, and 8,500 lb for the ice-activated system.

The average refrigeration load has been found to be 3.3 tons, and the average capacity of air conditioning systems is about 5.92 tons. The relation of load to capacity, $3.3 \div 5.92 = 0.56$ or 56 per cent, is that percentage of the time during the cooling season that the cooling equipment will be in operation. The average drawbar horsepower demand upon a locomotive, accordingly, consists of 56 per cent of the horsepower required for continuous operation and 44 per cent of the horsepower required for non-operation. Table 2 shows the drawbar horsepower that must be supplied by the locomotive for each air-conditioned car for continuous operation of the air conditioning system, for non-operation of the equipment, and for an average condition when the air conditioning equipment is operating continuously 56 per cent of the time and is not operating 44 per cent of the time. It is important not to overlook the horsepower demand on the locomotive when the air conditioning equipment is not operating, which includes the horsepower required to operate

the blower fan, to haul the weight of the equipment, to overcome drive and generator friction, and to replace the losses occasioned by the removal of current from the storage battery.

Fig. 2 shows the tractive resistance of a 75-ton passenger car with six wheel trucks without an axle generator, with a 4 kw generator load, and, for the same car with an increase in weight of 5 tons and a 20 kw axle

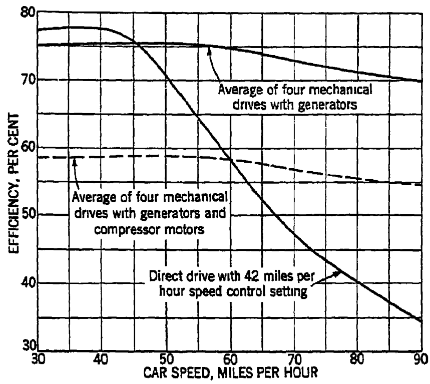

FIG. 1. EFFICIENCIES OF DRIVE MECHANISMS FOR RAILWAY AIR CONDITIONING SYSTEMS

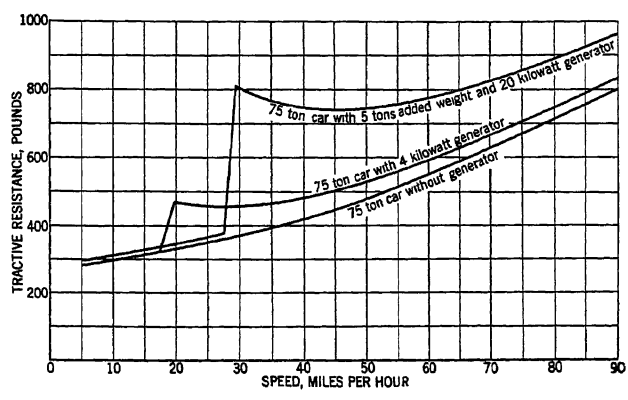

FIG. 2. TRACTIVE RESISTANCE OF 75 TON PASSENGER CAR WITH SIX WHEEL TRUCKS

generator load. The curve with the 4 kw generator is representative of a car before air conditioning, and the curve with the 20 kw generator and 5 tons added weight is representative of a car after air conditioning. At 50 mph, the tractive resistances of these two cars are 520 lb and 745 lb respectively, or a difference of 225 lb. Then: $\dfrac{225 \times 5280 \times 50}{60 \times 33000} = 29.7$ hp is required due to a 16 kw load and 5 tons added weight. Ten cars with a similar load would require 297 horsepower or roughly 10 per cent of the capacity of a 3,000 hp passenger locomotive.

Consideration must also be given to the power requirements for refrigeration while the car is standing or running at slow speeds. The electrical energy required for the ice-activated, steam, and internal combustion engine drive mechanical systems is easily supplied from the storage battery. Steam for the steam system can be supplied from the locomotive or from a stationary plant. The majority of the electro-mechanical systems are equipped with A. C.—D. C. motors. While standing in the yards and stations the A. C. motor is connected to a 220-volt, 3-phase circuit. The majority of these equipments are so ar-

TABLE 2. LOCOMOTIVE POWER DEMANDS FOR DIFFERENT AIR CONDITIONING SYSTEMS

System	Drawbar Horsepower per Car Required at Train Speeds			
	30 MPH	50 MPH	70 MPH	90 MPH
For Continuous Operation				
Electro-Mechanical	20.9	22.2	25.4	31.6
Direct Mechanical	16.8	19.5	29.5	42.6
Internal Combustion Engine Mechanical	3.4	4.6	7.0	11.9
Steam-Ejector[a]	7.6	9.3	12.5	19.0
Ice-Activated	3.2	4.5	6.8	11.6
For Non-Operation				
Electro-Mechanical	5.0	6.4	9.0	14.4
Direct Mechanical	5.2	6.3	8.7	13.5
Internal Combustion Engine Mechanical	2.2	3.5	5.8	10.6
Steam-Ejector	3.7	5.4	8.4	14.8
Ice-Activated	1.9	3.2	5.5	10.2
For Average Condition of 56 Per cent Continuous Operation and 44 Per cent Non-Operation				
Electro-Mechanical	13.9	15.2	18.2	24.0
Direct Mechanical	11.7	13.7	20.3	29.6
Internal Combustion Engine Mechanical	2.8	4.1	6.5	11.4
Steam-Ejector[a]	5.9	7.6	10.7	17.5
Ice-Activated	2.6	3.9	6.2	11.0

[a]In addition, steam is required from the locomotive to the extent of 230 lb per hour during the time the equipment is in operation.

ranged that, while operating on A. C. power, the D. C. motor may be used as a generator for battery charging. If an auxiliary circuit is not available the D. C. compressor motor may be operated from the storage battery for short periods of time. The direct drive mechanical compression systems are, also, equipped with A. C. motors for operation from auxiliary circuits. As these equipments can only be operated when connected to the auxiliary circuit or while the train is running above the cut in speed of the drive, many cars are equipped with an auxiliary hold-over system by which reserve cooling is available. Due to the characteristics of the direct drive, the air conditioning system operates at reduced capacity when the car is moving at speeds below 42 mph.

COST OF RAILWAY AIR CONDITIONING

The cost of railway air conditioning is usually expressed in terms of 1000 car-miles. The actual costs for the different systems, however, are dependent upon a number of variables. Based upon a survey of the most prominent railroads in air conditioning, and upon extensive tests, the values given in Table 3 are indicative of the present costs of air conditioning to the railroads.

TABLE 3. AIR CONDITIONING COSTS FOR RAILWAY AIR CONDITIONING

System	Gross Installation Cost	Costs per 1000 car-miles[a]			
		Fixed Charges	Maintenance Cost	Operation Cost	Total
Electro-Mechanical	$6,484.00	$ 8.65	$3.33	$0.99	$12.97
Direct Mechanical	8,515.00	11.35	2.33	0.93	14.61
Internal Combustion Engine Mechanical	5,750.00	7.67	3.30	1.99	12.96
Steam-Ejector	8,475.00	11.30	2.15	1.02	14.47
Ice-Activated	3,982.00	5.31	0.97	5.29	11.57

[a]For an average cooling season of 5 months, an average train speed of 50 mph and an average car mileage of 150,000 miles per year.

Gross Installation Cost

The gross installation cost, from which the fixed charges are derived, may be amortized on this basis:

1. Depreciation, at the rate of 12.5 per cent.
2. Interest, at the rate of 6 per cent.
3. Taxes and insurance at the rate of 1.5 per cent.

The fixed charges per 1000 car-miles for any type of system are:

$$FC = \frac{1000}{m}(0.20A) \tag{1}$$

where

FC = fixed charges, dollars per 1000 car-miles.
A = gross installation cost, dollars.
m = total number of car-miles traveled in one year.

Maintenance Cost

The average maintenance cost is based upon the experience of the railroads in maintaining several hundred air conditioning units. The maintenance cost per 1000 car-miles is:

$$MC = \frac{1000\,B}{m} \tag{2}$$

where

MC = maintenance cost, dollars per 1000 car-miles.
B = total annual maintenance cost, dollars.
m = total number of car-miles traveled in one year.

Operation Cost

The cost of operation is influenced by:

1. Speed of train operation.
2. Average drawbar horsepower required to operate air conditioning system.
3. Length of the cooling season.
4. Cost of power produced by the locomotive at $0.00493 per horsepower-hour.
5. Proportion of time the cooling equipment is operated during the cooling season considered as 56 per cent.
6. Cost of additional necessities as: *a*. Ice in bunker, at $4.42 per ton, *b*. Propane on the car, at $0.039 per pound, *c*. Steam at $0.021 per 100 pound.

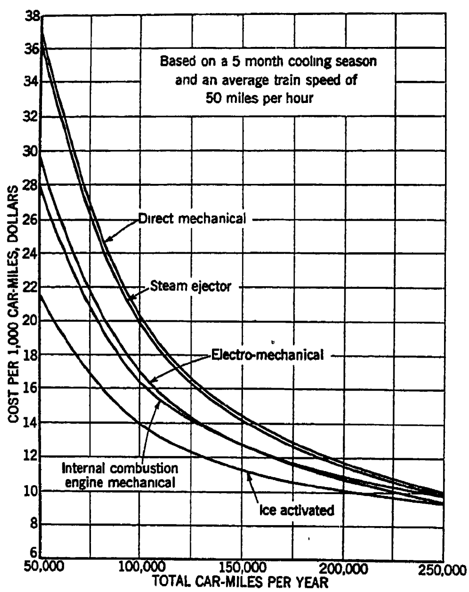

FIG. 3. COMPARATIVE TOTAL COSTS FOR RAILWAY PASSENGER CARS

The operation cost in dollars per 1000 car-miles is:

$$OC = \frac{1000\,K}{12\,S}\,(D \times E + 0.56\;F \times G) + \frac{1000\;(12 - K)}{12\,S}\,(H \times E) \qquad (3)$$

where

OC = operation cost, dollars per 1000 car-miles.
D = average drawbar horsepower demand on a locomotive at speed S.
E = cost per horsepower-hour, dollars.
F = additional necessities such as ice, steam or propane, pounds per hour.
G = cost of additional necessities, dollars per pound.
H = drawbar horsepower required when system is not operating.
S = speed of train operation, miles per hour.
K = length of cooling season, months.
0.56 = Proportion of operation time to total time during the cooling season.

Total Cost of Air Conditioning

When the fixed charges, maintenance cost, and operation cost are each expressed in terms of 1000 car-miles, addition of the three elements will give the total cost of air conditioning on that basis.

Comparisons of the total cost per 1000 car-miles for the five methods of air conditioning are shown in Fig. 3, representing costs for an average condition, namely a cooling season of five months and an average speed of 50 mph.

COOLING LOAD CALCULATIONS

The calculated heat gain for a railway passenger car is dependent on several variables which may be determined from the basic data given in Chapters 5, 6 and 8.

REFERENCES

Summary Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, November 24, 1936.

Engineering Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, April 15, 1937.

Report on Performance and Cost of Operation of 1937 Internal Combustion Engine Mechanical Compression Equipment for Air Conditioning Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, May 1, 1937.

PROBLEMS IN PRACTICE

1 ● What item is the greatest among the cooling loads figured in the design of a summer air conditioning system for a passenger car?

The heat from passengers.

2 ● To what extent does bright sunshine increase the cooling requirements of a passenger car?

About 1.2 tons of refrigeration.

3 ● What is the total refrigerating capacity generally required in a passenger car?

5.5 to 7 tons per car.

4 ● What is the effect of train speed upon the cooling requirements of a car?

Requirements are slightly increased because of increased heat transmission.

5 ● What is the fan capacity of the air conditioning unit in the average car?

2000 to 2500 cfm.

6 ● When is it economical to take all air for car cooling from outdoors?

When the outdoor wet-bulb temperature is lower than that in the car.

7 ● What various arrangements are used for distributing cooled air into cars?

Bulkhead delivery at center or ends of car, center duct, and side duct on one or both sides.

8 ● What types of cooling systems are used?

Ice-activated, steam-ejector, and mechanical compression systems.

9 ● What cooling medium is used for condensing the refrigerant in a railroad air conditioning system?

Outdoor air, sometimes with the aid of evaporative cooling.

10 ● How may adequate cooling of condensers be provided in hot desert regions?

By evaporative cooling with water sprays.

11 ● How is the temperature controlled in railroad cooling systems?

By intermittent operation of the compressor, the steam jet or the ice water circulating pump.

12 ● How much steam is required for car heating on the coldest days?

Pullmans 250 lb per hour, coaches 150 to 175 lb per hour, baggage cars 150 lb per hour.

13 ● At present costs which is likely to be the greater, fixed charges or operating costs?

The fixed charges for steam and mechanical compression systems, and operating costs for ice systems.

14 ● What would be the annual operating cost for a car equipped with an ice-activated system under the following conditions: a total mileage of 150,000 car-miles per year, a cooling season of 5 months, and an average train speed of 50 mph?

$1,705.00.

Chapter 33

INDUSTRIAL AIR CONDITIONING

Atmospheric Conditions Required, General Requirements, Classification of Problems, Control of Regain, Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reactions, Control Rate of Crystallization

IN the application of air conditioning to industrial processes, too much stress cannot be laid upon a thorough understanding by the air conditioning engineer of the problems involved. A complete knowledge of these problems is necessary before a satisfactory design can be made.

Individual processes and machines are changing rapidly and air conditions must be constantly revised to meet the new conditions.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process presents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions automatically. In departments where people are working, their health, comfort, and productive efficiency must be considered and often a compromise between the optimum conditions for processing and those required for the comfort of the worker is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 1 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
AUTOMOBILE	Assembly line	65	40
BAKING	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
BIOLOGICAL PRODUCTS	Vaccines	below 32	
	Antitoxins	38 to 42	
BREWING	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
CERAMIC	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
CHEMICAL	General storage	60 to 80	35 to 50
CONFECTIONERY	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
DISTILLERY	General manufacture	60	45
	Storage of grains	60	30 to 45
DRUG	Storage of powders and tablets	70 to 80	30 to 35
ELECTRICAL	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
FOOD	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35
FUR	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING (Continued)

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
INCUBATORS	Chicken	99 to 102	55 to 75
LABORATORY	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
LEATHER	Drying of hides	90	
LIBRARY	Book storage (see discussion in this chapter)	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCHES	Manufacturing	72 to 74	50
	Storage of matches	60	
MUNITIONS	Fuse loading	70	55
PAINT	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
PAPER	Binding, cutting, drying, folding, gluing	60 to 80	25 to 50
	Storage of paper	60 to 80	35 to 45
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	60 to 75	20 to 60
	Storage of rollers	60 to 80	35 to 45
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	60 to 70
	weaving	68 to 75	70 to 80
	Rayon— spinning	70	85
	twisting	70	65
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
TOBACCO	Cigar and cigarette making	70 to 75	55 to 65
	Softening	90	85
	Stemming or stripping	75 to 85	70

GENERAL REQUIREMENTS

In general, air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reaction; of warming or cooling to any desired temperature, and of providing ample air supply at all times. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum permissible temperature. Washing, purifying and recirculating of the air may be desirable. Good distribution is essential for the control of air motion and for the prevention of uneven conditions. Accurate, sensitive and reliable automatic control of humidity or temperature, or both, is vital in most cases.

Ordinarily, outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is used for driving the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes a normal degree of importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of over-all cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this book. A few of the salient points of the general subject are covered in this chapter.

CLASSIFICATION OF PROBLEMS

In general, any industrial air conditioning problem may be listed under one or more of the following four classes:

1. Control of Regain.
2. Control of Rate of Chemical Reactions.
3. Control of Rate of Biochemical Reactions.
4. Control of Rate of Crystallization.

CONTROL OF REGAIN

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance and general

quality of the product. This influence is due to the fact that the moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, come to equilibrium with the moisture of the surrounding air.

In industries where the physical properties of a product affect its value, the percentage of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

MOISTURE CONTENT AND REGAIN

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture. During the processing of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. A basis for calculating the regain of textiles is obtained by drying under standard conditions a sample from the lot and the dry weight thus obtained is used as a basis in the calculations to determine the regain.

The moisture content of an hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

CLASSIFICATION	MATERIAL	DESCRIPTION	RELATIVE HUMIDITY—PER CENT									AUTHORITY
			10	20	30	40	50	60	70	80	90	
Natural Textile Fibres	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fibre	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical qualities during manufacture. During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded.

These preliminary processes are carried out best in a relative humidity of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. To offset the extra work performed on the fiber as the spindle speed is increased, many cotton mills now carry 70 per cent relative humidity in the spinning rooms.[1] Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 57 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials release sensible heat equivalent to the latent heat of the moisture absorbed by the material, all of which may account for a large percentage of the total heat load.

CONDITIONING AND DRYING

In general, the exposure of materials to desirable conditions for treatment may be coincidental with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material.

When the final moisture content is lower than the initial one, the term *drying* is applied. If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco,

[1]The Present Status of Textile Regain Data, by A. E. Stacey, Jr. (*National Association of Cotton Manufacturers*, 1927).

for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content. Either conditioning or drying are frequently made continuous processes in which the material is conveyed through an elongated compartment by suitable means and subjected to controlled atmospheric conditions.

CONTROL OF RATE OF CHEMICAL REACTION

A typical example of the second general classification, that is, the control of the rate of chemical reactions, occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that during this process close control of both temperature and relative humidity should be maintained. Temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and gives a solution of known strength throughout the mercerizing period.

Another well-known example of this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding action on the rate of oxidization at the surface and allow the gases to escape as the chemical oxidizers *cure* the varnish film from the bottom. This produces a surface free from bubbles and a film homogeneous throughout.

Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

CONTROL OF RATE OF BIOCHEMICAL REACTIONS

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 65 per cent is maintained so as to hold the surface of the dough open to allow the CO_2 gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar process is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also come under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp of the fruit must be changed and the

skin cured and colored, after which the fruit is cooled to maintain as slow a rate of metabolism as possible. Ideal conditions range between 55 to 57 F and in no case should the temperature go below 49 F, as the starches then become fixed and are indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

CONTROL RATE OF CRYSTALLIZATION

The rate of cooling of a saturated solution determines the size of the crystals formed. Both temperature and relative humidity are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right temperature and relative humidity. If the cooling and drying is too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip through to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution, can the proper rate be obtained and a perfect coating assured.

The foregoing is presented as typical of a few of the problems met with in applying air conditioning to various industrial processes. They are far from complete but with the help of a few natural laws may assist in solving others where similar basic principles are involved.

CALCULATIONS

The methods for determining the proper heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 7 and 8. Because of the large number of motors and heat processing units usually prevalent in an industrial application, it is particularly important that operating allowances for the latent and sensible heat loads be definitely ascertained and used in the calculations to determine the total equivalent design load.

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PROBLEMS IN PRACTICE

1 ● Why is air conditioning required for some industrial processes?

To control the physical properties of the materials being processed.

Example. In the manufacture of chewing gum, it is rolled into slabs and scored. The scored slab must then be broken at the score marks to form the sticks. If the slab is too warm, breaking is impossible, if the slab is too cold or too dry, it becomes brittle and shatters, thereby producing much material to be reworked.

2 ● Why is it necessary to control the physical properties of the material being processed?

To permit permanent adjustment of machinery.

Example. In the manufacture of cigarettes, the amount of tobacco fed upon the paper tape is determined by pressure against springs. When the tobacco is over-moist and, therefore, over-soft, a great excess will go into the finished cigarette; when the tobacco is too dry and, therefore, harsh, too little goes into the finished cigarette.

3 ● A condition of 75 F dry-bulb temperature and 55 per cent relative humidity is being maintained in a cigarette manufacturing department. What will be the regain and moisture content of the tobacco?

$$\text{The regain, from Table 2} = 17.75 \text{ per cent.}$$

$$\text{The moisture content} = \frac{17.75 \times 100}{100 + 17.75} = 15.1 \text{ per cent.}$$

4 ● A 1-lb sample taken from a 100-lb batch of material is found to have a bone-dry weight of 0.89 lb. This material is to be processed under atmospheric conditions which should produce a regain of 15 per cent. Compute the finished weight for each original 100-lb batch.

Let W equal the number of pounds of moisture in a finished batch.

$$\frac{W}{89} = \text{regain} = 15 \text{ per cent} = \frac{15}{100}$$

$$W = 13.35$$

$$89 + 13.35 = 102.35 \text{ lb finished weight.}$$

5 ● A bundle of sea island cotton is found to have a bone-dry weight of 9.26 lb. What is the proper relative humidity at 75 F to produce a weight of 10 lb at equilibrium?

$$\begin{aligned} \text{Desired conditioned weight} &= 10.00 \text{ lb} \\ \text{Bone-dry weight} &= 9.26 \text{ lb} \\ \hline \text{Weight of moisture required} &= 0.74 \text{ lb} \end{aligned}$$

$$\text{Regain} = \frac{0.74}{9.26} \times 100 = 7.9 \text{ per cent.}$$

From Table 2, the proper relative humidity required is 60 per cent.

6 ● Compute the bone-dry weight of 1000 lb of manila rope which has been stored for a considerable period of time in a conditioned room at 75 F dry-bulb temperature and 50 per cent relative humidity.

Assuming that this material has come to equilibrium under the atmospheric conditions given, Table 2 shows a regain of 8.5 per cent.

Let W equal the total weight of moisture in pounds.

$1000 - W$ = bone-dry weight in pounds.

$$\frac{W}{1000 - W} = \text{regain} = 8.5 \text{ per cent} = \frac{8.5}{100}$$

$$W = 78.3 \text{ lb moisture}$$

$$1000 - 78.3 = 921.7 \text{ lb bone-dry weight.}$$

7 ● An egg evaporating plant wishes to dry 2000 lb of egg whites (85 per cent water) to crystalline form each 24 hours. The maximum permissible air delivery temperature in the dryer is 140 F. What air volume will be required, assuming that outside air is at 95 F dry-bulb and 78 F wet-bulb and that air leaves the dryer 70 per cent saturated?

Moisture to be removed = $2000 \times 0.85 = 1700$ lb. Using psychrometric chart and starting at the intersection of the vertical 95 F dry-bulb temperature line and the 46 per

cent humidity line, move horizontally to the right to the intersection with the 140] vertical temperature line at 13 per cent relative humidity; then move along the constan heat (or wet-bulb line) to its intersection with the 70 per cent relative humidity curv and read 97.5 F dry-bulb, which will be the temperature of the air leaving the dryer.

Moisture per cubic foot at 97.5 F and 70 per cent relative humidity	=	13.2 grains
Moisture per cubic foot at 95 F and 78 F wet-bulb	=	8.3 grains
Moisture added per cubic foot of air handled	=	4.9 grains

$$\frac{1700 \times 7000}{24 \times 60 \times 4.9} = 1685 \text{ cfm.}$$

No allowance is made for heat lost in the transmission to and from the dryer or for the heat required to raise the product from its entering temperature to that maintained in the dryer. This would necessitate a trial and error solution common to all drying problems.

Chapter 34

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, Design Procedure, Requirements for Suction and Velocity, Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Efficiency of Exhaust Systems, Selection of Fans and Motors, Corrosion

IN almost every industry some type of exhaust or collecting system is essential to achieve efficient and economical control of dusts and fumes. General design information is included in this chapter which is intended to relate primarily to factory exhaust systems.

CLASSIFICATION OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at all, owing to the rising air current carrying the dust up through the

breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN PROCEDURE

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[1]:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should not interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

[1]For more detailed requirements see Safe Practice Pamphlets Nos. 32 and 37, published by the *National Safety Council*, Chicago.

REQUIREMENTS FOR SUCTION AND VELOCITY

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

TABLE 1. SIZE OF CONNECTIONS FOR WOOD-WORKING MACHINERY

Type of Machine	Diameter of Connections in Inches
Circular saws, 12-in. diam.	4
Circular saws, 12-24-in. diam.	5
Circular saws, 24-40-in. diam.	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in.	5–6
With knives, 10-20 in.	6–8
With knives, 20-30 in.	6–10
Shapers, light work	4–5
Shapers, heavy work	8
Belt sander, belt less than 6 in. wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in.	5
Drum sander, 30 in.	6
Drum sander, 36 in.	7
Drum sander, 48 in.	8
Drum sander, over 48 in.	10
Disc sander, 24 in. diam.	5
Disc sander, 26-36 in. diam.	6
Disc sander, 36-48 in. diam.	7
Arm sander	4

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

DIAMETER OF WHEELS	MAX. GRINDING SURFACE SQ IN.	MIN. DIAM. OF BRANCH PIPES IN INCHES
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in. thick	43	3½
10 in. to 16 in., " " " 2 in. "	101	4
17 in. to 19 in., " " " 3 in. "	180	4½
20 in. to 24 in., " " " 4 in. "	302	5
25 in. to 30 in., " " " 5 in. "	472	6
Buffing—		
6 in. or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in. thick	57	4
13 in. to 16 in., " " " 2 in. "	101	4½
17 in. to 20 in., " " " 3 in. "	189	5
21 in. to 27 in., " " " 4 in. "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

TYPE OF INSTALLATION	STATIC SUCTION IN INCHES OF WATER
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2
Fur and felt machinery exhaust	2–4
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2–3
Conveying bulky and heavy materials	2
	3–5

than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al[2] give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis[3]:

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \tag{1}$$

where

V = velocity at point, feet per minute.
A = area of opening, square feet.
x = distance along axis, feet.
Q = volume of air handled, cubic feet per minute.

Velocity Contours

It is possible by use of a specially constructed pitot-tube[4] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[5].

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[2]Control of the Silicosis Hazard in the Hard Rock Industries. I A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P. Choate. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

[3]The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October 1933).

[4]Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (*A.S.M.E. Transactions*, Vol. 54, 1932).

[5]Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 387).

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

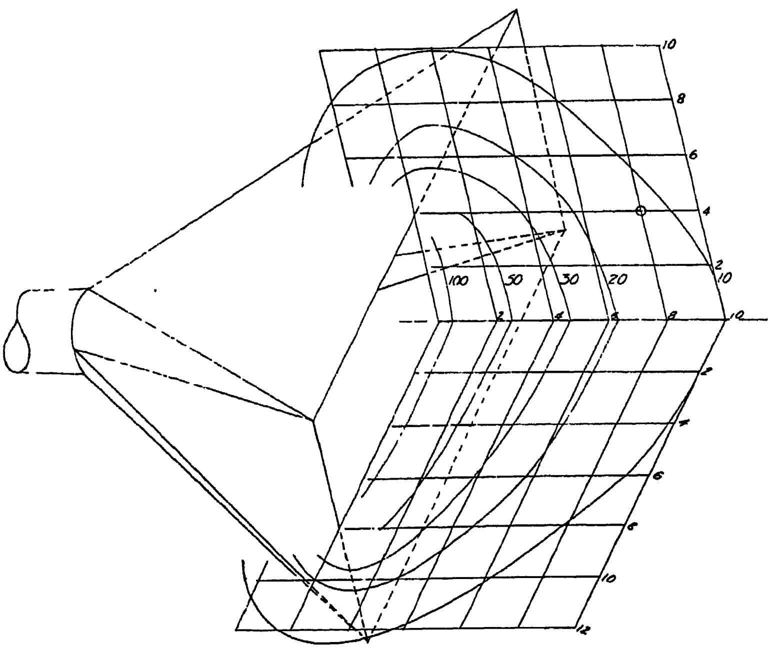


FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

Air Flow from Static Readings

The volume of air flow through any hood may be determined from the following equation:

$$Q = 4005 f A \sqrt{h_t} \tag{2}$$

where

Q = volume of air flow, cubic feet per minute.

A = area of connecting duct, square feet.

h_t = static suction at throat of hood, inches of water.

f = orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_t}} \tag{3}$$

where h_v is the velocity head in the connecting duct.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity at any point along the axis varies inversely as the area of the opening and the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be subdivided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12-in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \tag{4}$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating[6], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 2 in. wide and for effective

[6]Health Hazards in Chromium Plating, by J. J. Bloomfield and Wm. Blum (*U. S. Public Health Report*, Vol. 43, No. 26, September 7, 1928).

ventilation a 2,000 fpm exhaust air velocity at the slot face is advisable. In addition, the duct should not be required to draw the air laterally for a distance of more than 18 in. and the level of the solution should be kept 6 to 8 in. below the top of the tanks.

Flexible Exhaust Systems

The flexible exhaust tube method may be advantageously used for removing dust or fumes. Flexible tubes having one end connected to an exhaust system and a slotted hood attached to the other end may be shaped at will to fit in with industrial processes without affecting the ease of operation. Efficient dust or fume removal may be had with use of relatively small exhaust volumes. This type of system may be used on swing grinders, portable grinding wheels, soldering operations, stone cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept below 100 parts per million. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located in a horizontal position slightly below the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth, and the vapors may be discharged through a suitable stack to permit dilution, but it is better practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray fume removal[7].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is opened to its maximum height.

DUCT SYSTEM DESIGN

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and par-

[7]For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926, Harrisburg, Pa.

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust	2000
Wood chips and shavings	3000
Sawdust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5a and 5b may be used as tests to determine the conveying efficiency of a system[8]. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:

$$V = 13{,}300 \frac{s}{s+1} d^{0.57} \tag{5a}$$

For horizontal ducts:

$$V = 6000 \frac{s}{s+1} d^{0.40} \tag{5b}$$

where

V = air velocity in duct, feet per minute.

s = specific gravity of particles.

d = average diameter of largest particles conveyed, inches.

Example 2. Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog (0.57 × log 0.37) = 0.568; the required velocity is, therefore, 5500 fpm.

[8]Determining Minimum Air Velocities for Exhaust Systems, by J. M. DallaValle (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1932, p. 639).

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

Elbow Center Line Radius in Per Cent of Pipe Diameter	Loss in Per Cent of Velocity Head
50	75
100	26
150	17
200 to 300	14

Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 3, Chapter 29. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \tag{6}$$

where

h_c = the pressure drop through the cyclone, inches of water.
V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts,

feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

For additional information on Dust and Cinders, see Chapter 26, Air Cleaning Devices.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[9].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level. Commonly accepted values of threshold limits for the usual gases and vapors are given in Table 7.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For

TABLE 7. THRESHOLD LIMITS OF COMMON VAPORS AND GASES[a]

SUBSTANCE	SPEC. GRAV. OF GAS OR VAPOR (AIR 1)	INFLAMMABLE LIMITS (%)	PHYSIOLOGICAL ACTION	MAXIMUM ALLOWABLE CONCENTRATION (PPM)
Chlorine	2.486	non-inflamm.	irritant	0.35
Ozone	5.5	do	do	0.80
Hydrogen chloride	1.2678	do	do	10.0
Sulphur dioxide	2.2638	do	do	10.0
Carbon monoxide	0.9671	12.5–74	asphyxiant	100.0
Hydrogen sulphide	1.190	4.3–46	do	85–130
Benzene	2.73	1.4–7.0	anesthetic	100.0
Methanol	1.1	7.5–26.5	do	100.0
Carbon tetrachloride	5.3	non-inflamm.	do	100.0

[a]The Prevention of Occupational Diseases, by R. R. Sayers and J. M. DallaValle (*Mechanical Engineering*, Vol. 57, No. 4, April, 1935).

particular features concerning special fans, consult the *Catalog Data Section* of THE GUIDE and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapters 27 and 38.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to

[9]Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 353).

TABLE 8. MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION[a]

Type of Fume Conveyed	Protective Material to be Used
Chlorine	Rubber lining or chrome-nickel alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickel alloys
Hydrochloric acid	Rubber lining, chrome-nickel alloys
Nitrous gases	Nickel-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol. 24, 1932).

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 8. Hoods and ducts when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

PROBLEMS IN PRACTICE

1 ● What is the most common method of reducing total air volumes handled in cases employing large hoods over apparatus covering a large area?

The use of the petticoats on large hoods which permits a comparatively high air velocity at the rim of the hood and controllably small velocities in the center.

2 ● Why is it not permissible to connect up emery wheels and buffing wheels to the same exhaust system?

Emery wheels and buffing wheels should be handled by separate systems because of the fire hazard, as it is possible for sparks from the emery wheels to ignite the lint and dust from the buffing wheels when both are carried through the same system.

3 ● A tank, 4 ft by 8 ft, contains a fluid which gives off injurious vapors. A large hood is located 30 in. above the top of the tank and extends slightly over its edges. Assuming that a velocity of 60 fpm is required to adequately control the vapors near the edges of the tank, calculate the air flow required.

Using Equation 4, $P = 2 \times 4 + 2 \times 8 = 24$ ft; $D = 30$ in. $= 2.5$ ft; $V = 60$ fpm.

$$\text{Hence, } Q = 1.4 \times 24 \times 2.5 \times 60 = 5040 \text{ cfm.}$$

4 ● Silica dust with a specific gravity of 2.65 is being conveyed in a duct system. The velocity measured in a vertical portion of the system is found to be 2700 fpm. What is the maximum diameter particle transported at this velocity?

$$\text{Using Equation 5a, } 2700 = 13{,}300 \times \frac{2.65}{3.65} \times d^{0.570}$$

from which

$$d = (0.28)^{1.75} = 0.11 \text{ in.}$$

5 ● What special materials may be used to resist chemical corrosion in a system exhausting gases and fumes?

Various protective materials are available for exhaust systems depending largely upon the type of fumes conveyed. Nickel-chrome alloys, aluminum coated metals and rubber linings are extensively used. Also protective rubberized paints are available which may be applied for conveying chlorine and hydrochloric acid fumes.

Chapter 35

DRYING SYSTEMS

Drying Methods, Driers, Mechanism of Drying, Moisture, General Rules for Drying, Equipment, Humidity Chart, Combustion, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation.

DRYING METHODS

Drying may be accomplished in any one or combination of the following methods:

1. Radiation.
2. Conduction, or direct contact.
3. Convection.

Radiation

The source of heat for radiation may be either the sun, or heated surfaces. Sun drying is practiced where danger from rain is slight, and where sufficient time can be allowed. Where a strict adherence to a schedule is necessary, or where dusty atmosphere is present, this method is not in favor. Fruits are often dried in the sun.

Radiation from hot surfaces (heated by steam, electricity, or other means) furnishes generally, from one-third to one-half the total heat required for evaporation. Convection currents set up by these hot surfaces and the cooler materials carry the balance of the heat.

TABLE 1. DRIERS FOR EVAPORATION OF WATER

Type	Kind	Materials Handled	Means of Handling	Temp. Range Deg F	Heat Supply	Uses and Remarks
Batch or Intermittent	Compartment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray, Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For High Production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

Conduction or Direct Contact Drying

This method of drying is advantageous where the material can be flowed on to the drying surface and the dried material scraped off, or where the material to be dried can be handled in a sheet, and where there is no danger of subjecting the product to the full temperature of the heating medium. The source of heat for this method may be steam, electricity, hot oil or hot water.

Convection

The circulation of heated air or other gases about the material to be dried is generally termed convection drying. The convection may be either natural or forced. With forced circulation, the temperature of the

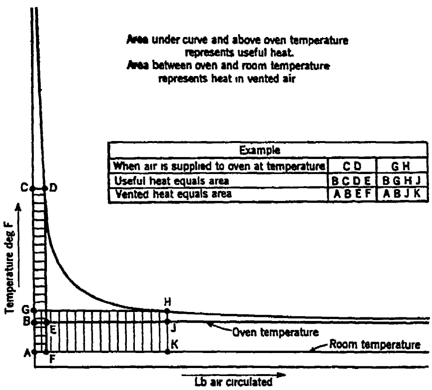


FIG. 1. RELATION BETWEEN USEFUL AND TOTAL HEAT SUPPLIED

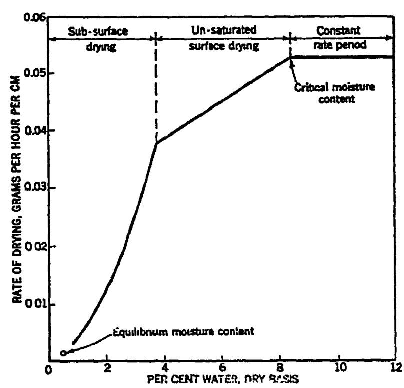


FIG. 2. RATE OF DRYING OF WHITING SLAB

drier is more uniform and the rate of drying is much higher than with natural circulation. Where humidity is used, the control is much easier, and more accurate.

The source of heat for a convection drier may be steam, electricity, hot water, oil-fired heater, gas-fired heater, or coal furnace.• Where either oil, gas or coal is used, the type of heater may be direct or indirect; *i.e.*, the products of combustion may be used (direct), or the circulated air may be heated through an interchanger (indirect).

Where the direct type is used, there is naturally a higher thermal efficiency, but it can only be used where the odor, soot, or the chemical elements of the products of combustion do not affect the material being dried. When heat economy is an important consideration this method (Fig. 1) may be used, permitting a small amount of air to be circulated, if a sacrifice of accurate control of temperature and humidity can be justified.

DRIERS

The term *adiabatic drier* is applied to a drier in which all the heat is supplied by air externally heated. The temperature of the air in the

drier decreases as a transfer of heat to the material being dried takes place. Where part or all of the heat is supplied by steam coils or other means, within the drier itself, the drier is known as a *constant temperature drier.* Driers using little air for heating medium with a high temperature drop, are difficult to hold at uniform temperatures; the more air used, the easier it is to secure accurate control of temperature and humidity. Driers may be classified as shown in Table 1.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period.
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material assumes a temperature corresponding to the wet-bulb temperature of the surrounding air, or slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content.*

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or sub-surface drying.

As drying progresses another point called *equilibrium moisture content* is reached, where the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal, and drying ceases. The drying of a slab of whiting is shown in Fig. 2 and illustrates the principles pointed out above. The factors affecting the variations of drying rates during the above periods are pointed out in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the

intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the

TABLE 2. FACTORS INFLUENCING DRYING

Factor	Drying Period	
	Constant Rate, Unsaturated Surface	Sub-Surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate, because with decreased viscosity, diffusion increases
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached; drying then ceases
Air Velocity	Drying rate varies approximately as the 0.6 power of the velocity	No effect
Air direction	Drying rate increases the more nearly the air blows perpendicular to surface; for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

falling rate period being confined solely in practice, to unsaturated surface drying.

MOISTURE

Moisture in the solid may be in either of two forms:

1. Capillary or free.
2. Hygroscopic or chemically combined.

Free moisture is contained in the capillary spaces between the particles or fibers of the materials. The loss of this moisture changes only the weight of the material. *Chemically combined* or hygroscopic moisture is intimately associated with the physical nature of the material and its removal changes both the physical characteristics as well as the chemical properties. The amount of hygroscopic moisture a material can contain is limited. This limit is called the *fiber saturation point*. When material is dried below this point, care must be exercised to avoid physical changes in the material, such as shrinkage, hardening, etc. All hygroscopic materials have definite equilibrium moisture contents dependent on temperature and humidity. Materials are frequently dried to a lower moisture content than those of equilibrium conditions in use, and allowed to regain the necessary moisture after leaving the drier to equalize the moisture in the material. Fig. 3[1] shows the equilibrium moisture content of wood.

[1] *U. S. Department of Agriculture Bulletin*, No. 1136.

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of fast drying and smaller requirements for ventilation. The amount of moistu that can be carried by a pound of air increases rapidly with rise in te perature as shown in the humidity chart of Fig. 4. Too high a temper ture may cause spoilage of materials; many materials calcine or chan their chemical properties if heated too hot; gypsum and glauber salts lo some of the chemically combined water, fall apart, and change th chemical properties. Too high or rapid rise in temperatures in dryi lumber or ceramics may create a liquid vapor tension within the materi so high that the cells explode, causing permanent injury to the fiber. too high a temperature is used on some chemicals, they begin to rea

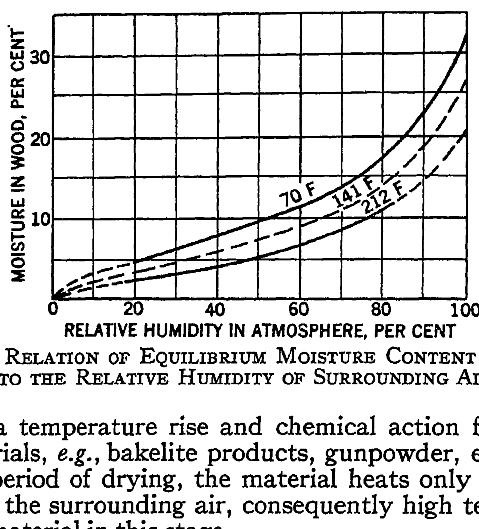


FIG. 3. RELATION OF EQUILIBRIUM MOISTURE CONTENT IN WOOD TO THE RELATIVE HUMIDITY OF SURROUNDING AIR

exothermally; a temperature rise and chemical action from within wi burn the materials, *e.g.*, bakelite products, gunpowder, etc. During th constant rate period of drying, the material heats only to the wet-bul temperature of the surrounding air, consequently high temperatures wil not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many material tend to case-harden, dry on the outside, forming a skin which retards th moisture flow from the inside to the surface, or stops it completely, and so increases the drying time very much or causes a change of the physica properties of the material. It is often necessary to add humidity to th air in the initial stage of drying. Lumber case-hardens, cracks, anc warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp while others crack i not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures

below 120 to 140 F are used the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the drier during periods of high atmospheric humidity; where a high degree of uniformity is required it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Complete descriptions of such dehumidifying systems are given in Chapter 24 on Cooling and Dehumidification Methods.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous driers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air even with the use of filters, may cause dust already in the drier, to be blown against the material, ruining the finish. Table 3 presents data on drying of various materials.

EQUIPMENT FOR DRYING

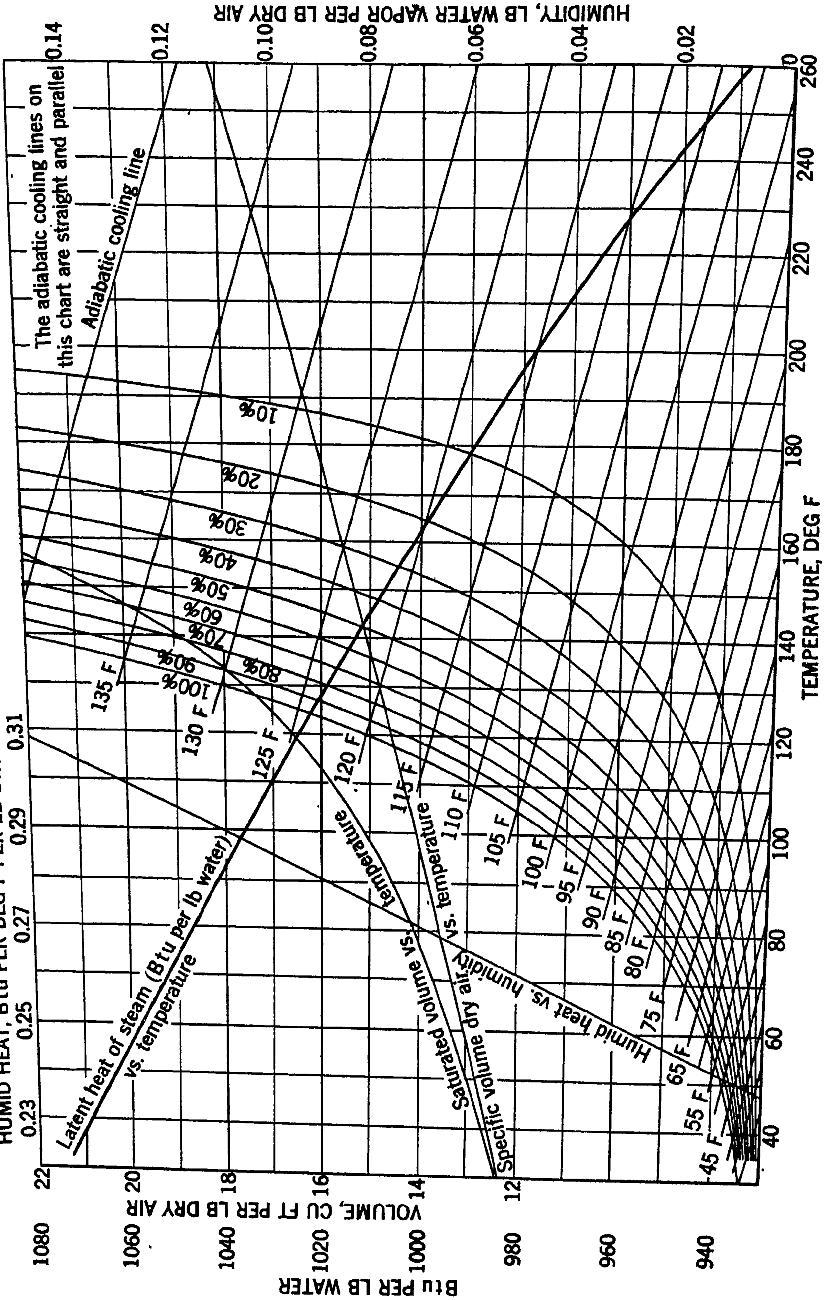
Equipment for drying may be divided into the following classes:

1. Heat and humidity supply.
2. Methods of handling.
3. Ovens.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct-fired equipment care must be used in the selection of burners and sufficient combustion space allowed to insure complete combustion of fuel. Humidity can be obtained in driers by the use of steam spray, humidifiers, or recirculation.

Methods of *handling of material* have been indicated in Table 1.

For low temperature work up to 200 F *ovens* and driers are commonly built of two thicknesses of insulating board (fireproof preferred), with air space between. As the temperature increases materials better able to


HUMIDITY, LB WATER VAPOR PER LB DRY AIR
0.14
0.12
0.10
0.08
0.06
0.04
0.02
0
TEMPERATURE, DEG F
40
60
80
100
120
140
160
180
200
220
240
260
HUMID HEAT, Btu PER DEG F PER LB DRY AIR
0.23
0.25
0.27
0.29
0.31
VOLUME, CU FT PER LB DRY AIR
22
20
18
16
14
12
Btu PER LB WATER
1080
1060
1040
1020
1000
980
960
940
The adiabatic cooling lines on this chart are straight and parallel
Adiabatic cooling line
10%
20%
30%
40%
50%
60%
70%
80%
90%
100%
135 F
130 F
125 F
120 F
115 F
110 F
105 F
100 F
95 F
90 F
85 F
80 F
75 F
65 F
55 F
45 F
Latent heat of steam (Btu per lb water) vs. temperature
Saturated volume vs. temperature
Specific volume dry air vs. temperature
Humid heat vs. humidity

withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature driers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous drier where the ends are open, heat and air leakage becomes important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled See Figs. 5 and 6.

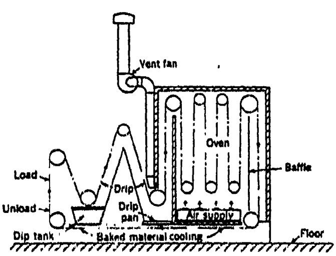


FIG. 5. SMALL PART MULTIPLE PASS OVEN

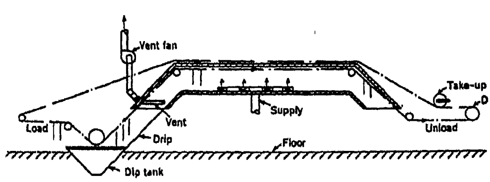


FIG. 6. INCLINED END ENAMELING OVE

HUMIDITY CHART FOR DRYING WORK

In drying problems the chemical engineer uses different psychrometr values than those used by the heating, ventilating and air conditionin engineer. The humidity chart illustrated in Fig. 4 is based upon valu determined from the following explanations:

Humidity (*H*) is the number of pounds of water vapor carried by o pound of dry air.

Percentage Humidity (*%H*) is the number of pounds of water vap carried by one pound of dry air at a definite temperature, divided by t number of pounds of vapor that one pound of dry air would carry if were completely saturated at the same temperature.

Per Cent Relative Humidity (Φ) is the ratio of weight of water vap contained in any given volume of air, to the weight of water vapor prese in the same volume of saturated air, all values referring to the sar temperature.

To convert from one relation to the other,

$$\% H = \frac{29.92 - p_s}{29.92 - p} \times \Phi$$

where

p_s = vapor pressure of water, inches mercury; at dry-bulb temperature, degr Fahrenheit.

$p = \Phi\, p_s$.

COMBUSTION

Where products of combustion are used directly in the oven, a knc ledge of their formation and heat values is important. The properties

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Apples	140–180		6 Hrs
Armatures Varnish	200		2.5 Hrs
Banana Food ¼ in. Thick	140		4–6 Hrs
Barrels	300		15 Min
Beans	140		18 Hrs
Bedding	150–190		
Blankets	120		40 Min
Brake Lining	325		12 Min
Brick continuous	350 to 90		24 Hrs
Briquets	1100		108 Min
Cabbage Raw	150		4.5 Hrs
Candied Peel	165		2 Hrs
Casein	180		5 Hrs
Cereals	110–150		
Ceramics before firing	150	70 to 20	24 Hrs
Chicle	95–100		
Coco-fiber mats	170–210		10 Hrs
Cocoanut	150–200		4–6 Hrs
Coffee	160–180		24 Hrs
Conduit (Enamel)	400 Max		2 Hrs
Cores, Oil sand for molding ½—1 in. thick	300		30 Min
Black sand with goulic binder about 0.6 of time 3 in. thick	480		2.5 Hrs
8 in. thick	480		4.5 Hrs
16 in. thick	700		10 Hrs
Cores, Crank case (in continuous ovens)	525–600		2–3 Hrs
Cores, Radiator (in continuous ovens)	275–450		1.5 Hrs
Cornstalk Board	150		2 Hrs
Cotton Linters	180		
Enamels synthetic			
Finish coat on autos	225		2 Hrs + Air Dry
Ice boxes all metal (white)	290–425		1 Hr
Ice boxes wood inside (white)	225		3 Hrs
Enamel not synthetic			
Fence posts green	200		1 Hr
Golf balls (white)	90–95	40–50	18–36 Hrs
Small parts (auto) black	450		1 Hr
Steel furniture	225–300		30–350 Min
License plates	250		1.5 Hrs
Feathers	150–180		
Films, Photographic	85–110		20–30 Min
Fruits and Vegetables	140		2–6
Furs	110		
Gelatin	110		
Glue bone, thin sheets on wire trays	70–90		6–9 Days
Glue skin	70–90		2 Days
Glue size on furniture	130		4 Hrs
Gut	150		
Gypsum board ⅜ in. thick Start Wet	350		60 Min
Finish	275		
Gypsum block	350–190		8–16 Hrs
Hair felt	180–200		
Hair goods	150–190		1 Hr
Hanks on poles	120		2 Hrs
Hats felt	140–180		
Hides thin leather	90		2–4 Hrs
Hides heavy	70–90		4–6 Days

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Hops	120–180		
Ink printing	70–300		
Japan beds	300		1.5–2 Hrs
Japan cash register	300–450		1.5 Hrs
Japan metal shelving	200		30 Min
Knitted fabrics	140–180		
Leather mulling	78–95	85	
Leather thick sole	90	70	4 Days
Leather uppers	80		2–3 Days
Linoleum varnish	110–145	10–30	6–10 Hrs
Lithographing on tin color work	250–270		18–25 Min
Lithographing on tin Japan	350		
Lumber green hardwood	100–180		3–180 Days
Lumber green soft wood	160–220		2–14 Days
Macaroni	90–110		7.5–8 Hrs
Matches	140–180		
Matrix	350		15 Min
Milk and other liquid foods spray dried	135–300		Instantaneous
Millboard sheets	95		10 Hrs
Moulds green sand C.I. flasks (one surface only exposed) 8 in. thick	600		6 Hrs
Moulds green sand C.I. flasks (one surface only exposed) 13 in. thick	700		13 Hrs
Motors, field coils	180		6 Hrs
Motors, stators	250		6.5 Hrs
Noodles	90–95		
Nuts	75–140		24 Hrs
Oil cloth	150		
Paint, wood wheels	150	35	8–24 Hrs
Paint, on sheet metal	350–140	22–30	2.5 Hrs
Paper, machine dried	180		
Paper, air dried	90–200		
Paper wall, ground coat	140		3 Min
Paper wall, varnished	140–160	45	15 Min
Paper cardboard, spirit varnish	150		1–2 Min
Peaches	135		26 Hrs
Pears	140		24 Hrs
Peas	150		6 Hrs
Potatoes sliced	85		4 Hrs
Potatoes steamed	170		6.5 Hrs
Prunes	140		
Rags	180		
Ramie fiber	140		10 Hrs
Rice	150		
Rock wool insulation	300		8 Hrs
Rubber	85–90		6–12 Hrs
Rubber reclaimed	140–200		1–2 Hrs
Rugs	190		4–8 Hrs
Salt	350		Rotary Drier
Sand loose 1 in. deep	300		10–15 Min
Sausage casings	110		5 Hrs
Shade cloth	240		1–2 Hrs
Shirts	120		20 Min
Soap	100–125		12–72 Hrs
Starch	180–200		1–4 Hrs
Stock feed mixed	180–220		20–30 Min
Storage battery plates	100–110	90 for	24 Hrs
	250	Low for	6 Hrs
Sugar	150–200		20–30 Min

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

MATERIAL	TEMPERATURE DEG F	PER CENT RELATIVE HUMIDITY	DRYING TIME
Tanin and other chemicals (spray dried)	250–300		Instantaneous
Terra Cotta (air drying in conditioned room)	150–200		12–96 Hrs
Tobacco leaves	85–130		12 Hrs
Tobacco stems	180–200		12 Hrs
Varnish refrigerator boxes	110	35	5–7 Hrs
Varnish steering wheels	110–140	25–35	Overnight
Veneer ¼ in. 3-ply	120–130	35–40	6–8 Hrs + 2 Hrs acclimation
1³⁄₁₆ in. 5-ply	120–130	35–40	16–18 Hrs + 4 Hrs acclimation
1¼ in. 5-ply	120–130	35–40	20–24 Hrs + 4 Hrs acclimation
Vitreous Enamel sheets before firing	170		
Wallboard pasted plywood	300		15–20 Min
Wallboard fiber insulating, roller type drier	300–385		2½–3 Hrs
Wallboard fiber insulating, truck type drier	300–385		24–48 Hrs
Walnuts	100		24 Hrs
Wheat, corn, oats, rice, barley	180		
Wire cloth Japan	200		20 Min
Wool	105		

[a]See references at end of chapter.

the common constituents of fuel are shown in Table 4. The heating values of oils are shown in Fig. 7. The sensible heat in Btu contained in the products of combustion of an average fuel oil and various gases is given in Fig. 8. The problem of securing complete combustion in a heater is important, in order to secure efficiency and the absence of soot formation, but unlike the ordinary power or heating boiler, excess air need not be maintained at a minimum in most cases. Excess air is generally admitted either in the heater or before the products go into the drier.

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the temperatures, humidities, and velocities to be actually used in the full sized drier, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of drying calculations:

H = humidity of air, pounds of water vapor per pound of dry air.
G = pounds of dry air supplied to the drier per unit of time.
S = pounds of stock dried per unit of time in a continuous drier.
S' = pounds of stock charged per batch to a discontinuous drier.
Θ = time.
Q = total heat supplied to the drier.

t = air temperature.
t' = stock temperature.
t'' = average stock temperature over short time interval, in a batch drier.
t_w = wet-bulb temperature.
s' = specific heat of the stock.
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock.
r = heat of evaporation of water.
s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters and (2) where it leaves the drier.

Air driers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the drier as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the drying equipment itself*. In any continuously operating drier of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S(w_1 - w_2) \qquad (2)$$

TABLE 4. GAS COMBUSTION CONSTANTS[a]

Gas	Chemical Formula	Molecular Weight	Cu Ft per Lb	Heat of Combustion, Btu per Lb, Gross	Heat of Combustion, Btu per Lb, Net	Lbs per Lb of Combustible, Required for Combustion, O_2	Required for Combustion, N_2	Required for Combustion, Air	Flue Products, CO_2	Flue Products, H_2O	Flue Products, N_2
Carbon	C	12.000	....	14,140	14,140	2.667	8.873	11.540	3.667	....	8.873
Hydrogen	H_2	2.015	187.723	61,100	51,643	7.939	26.414	34.353	....	8.939	26.414
Oxygen	O_2	32.000	11.819	....	....	....	....	....	....	....	....
Nitrogen	N_2	28.016	13.443	....	....	....	....	....	....	....	....
Carbon Monoxide	CO	28.000	13.506	4,369	4,369	0.571	1.900	2.471	1.571	....	1.900
Carbon Dioxide	CO_2	44.000	8.548	....	....	....	....	....	....	....	....
Methane	CH_4	16.031	23.565	23,912	21,533	3.992	13.282	17.274	2.745	2.248	13.282
Ethane	C_2H_6	30.046	12.455	22,215	20,312	3.728	12.404	16.132	2.929	1.799	12.404
Propane	C_3H_8	44.062	8.365	21,564	19,834	3.631	12.081	15.712	2.996	1.635	12.081
Sulphur Dioxide	SO_2	64.060	5.770	....	....	....	....	....	....	....	....
Water Vapor	H_2O	18.015	21.017	....	....	....	....	....	....	....	....
Air	....	28.900	13.063	....	....	....	....	....	....	....	....

[a]All gas volumes corrected to 60 F and 30 in. mercury barometric pressure dry.

In discontinuous driers, *e.g.*, compartment driers, the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S^1 \frac{dw}{d\Theta} \tag{2a}$$

In the continuous drier, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)\,(H_2 - H_1) + S(t'_2 - t'_1)\,(s^1 + w_1) + B \tag{3}$$

Equation (3) assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average

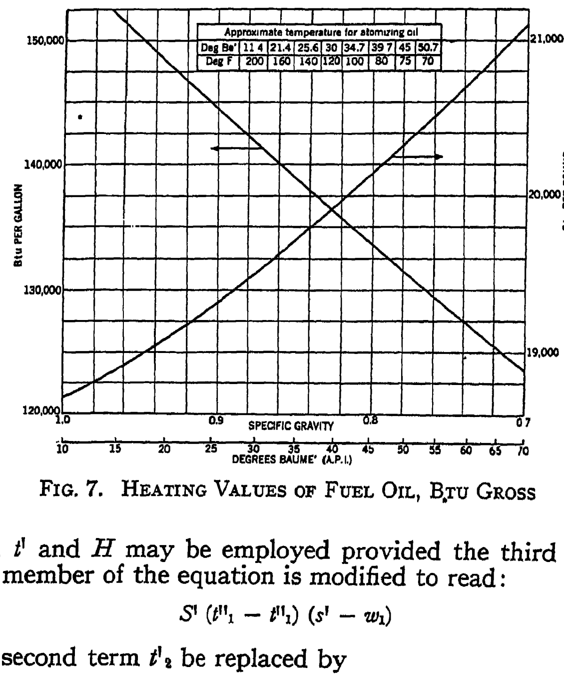


FIG. 7. HEATING VALUES OF FUEL OIL, BTU GROSS

values of t, t^1 and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S^1\,(t''_1 - t''_1)\,(s^1 - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically the error introduced by using a small number of long periods and employing average values of the variables over each, rarely introduces serious error. The evaluation of equation (2a) may be approximated in a similar manner.

The first term of the right hand member of equation (3) represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air

into contact with the stock with sufficient intimacy so that the air leaving the drier is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air, due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air is imperative.

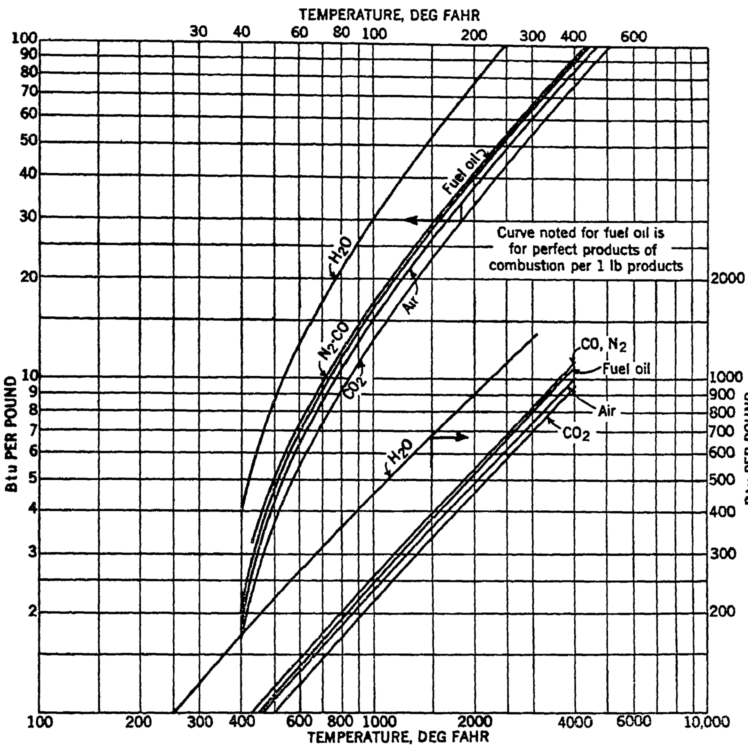


FIG. 8. HEAT CONTENT OF GASES ABOVE 32 F IN BTU PER POUND

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be

brought up to 50 per cent humidity at 150 F. The drier is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current drier will be employed and the air in this drier will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in

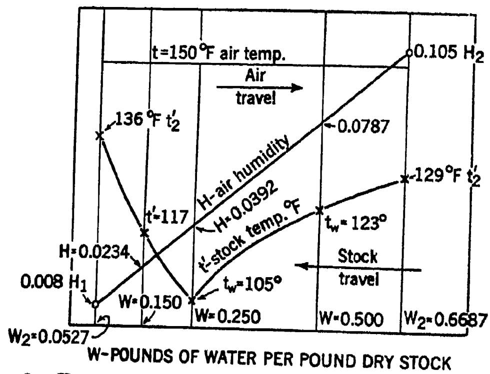


FIG. 9. TEMPERATURE-HUMIDITY RELATIONS IN A DRIER

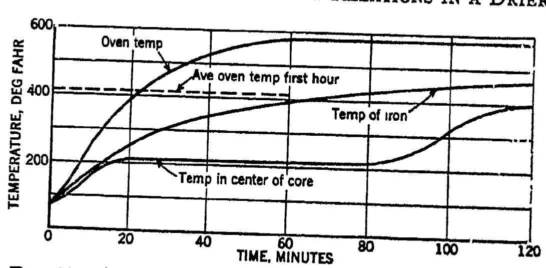


FIG. 10. CORE DRYING TIME TEMPERATURE RELATIONS

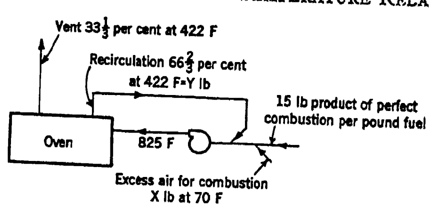


FIG. 11. CORE DRYING DIAGRAM OF COMBUSTION PRODUCTS AND AIR

contact, and is found experimentally to maintain wet-bulb temperature until the moisture content has fallen to 20 per cent. From this point its temperature rises progressively as it dries. In this range the difference in temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667: \; w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the drier is 50 per cent saturated at 150 F from Fig. 4, $H_2 = 0.105$. Similarly, $H_1 = 0.008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0.097$ lb water evaporated per pound dry air.

Inspection of equation (2) shows that (H) is linear in w. Hence, one can construct on Fig. 9, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the drier has a temperature of 150 F and a humidity of 0.105, Fig. 4 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 9. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of equation (2) or by reading directly from the diagram, the value being 0.0392. Fig. 4 shows that the corresponding wet-bulb temperature is 105 F. Any intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 9.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from equation (2)) and from Fig. 4, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F. The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t^1}{t - t_w} = \frac{w}{0.25}$$

At the point $w = 0.15$, $\Delta t^1 = 33$ F, $t^1 = 117$ F. The temperature of the stock leaving the drier, similarly computed, is 136 F.

Fig. 9 thus computed gives in graphical form the information as to the temperature humidity relationships in the drier. The air requirements can be computed by equation (2). Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 4 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per pound; 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the drier is 11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the drier, as a whole, or to any section of it, may be computed from equation (3).

High Temperature Drier

In the design of a high temperature drier unit a method of approach to the necessary calculations involved are outlined as follows:

Example 1. Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface having an average heat transfer of 0.3 Btu per square foot per degree per hour. Drying time as determined by test is 2 hr (Fig. 10). Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the drier at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. Fuel oil of 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6.75 lb per gallon and having 15 lb product per pound fuel

for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution. Heat required per ton of cores:

	Lb Material × Temp. Rise × Sp. Ht.	=	Btu
Sand	0.91 × 2,000 × (400 − 70) × 0.2	=	120,120
Binder	0.03 × 2,000 × (400 − 70) × 0.4	=	7,920
Water heating	0.06 × 2,000 × (212 − 70) × 1.0	=	17,040
Water evaporation	0.06 × 2,000 × 970 (Fig. 4)	=	116,520
Water superheating (approx. 50 per cent reaches 575 F)	= 0.5 × 0.06 × 2,000 × (575 − 212) × 0.45	=	9,800
Total Heat			271,400 Btu

HEATING LOAD FIRST HOUR

	HEATED TO			BTU
Sand	212 F	$\frac{142}{330}$ × 120,120	=	51,688
Binder[a]	212 F	$\frac{142}{330}$ × 7,920	=	3,408
Water	212 F		=	17,040
Evaporation	66.7%	0.667 × 116,520	=	77,680
Superheat	66.7%	0.667 × 9,800	=	6,530
Total Per Ton				156,346
For 6 ton		6 × 156,346	=	938,076
Steel plates	390 F	320 × 30,000 × 0.12	=	1,152,000
Radiation[b]	422 F Avg.	352 × 856 × 0.30	=	90,394
Total				2,180,470

HEATING LOAD SECOND HOUR

Sand	400 F	$\frac{188}{330}$ × 120,120	=	68,432
Binder[a]	400 F	$\frac{188}{330}$ × 7,920	=	4,512
Water				
Evaporation	33.3%	0.333 × 116,520	=	38,840
Superheat	33.3%	0.333 × 9,800	=	3,270
Total Per Ton				115,054
For 6 ton		6 × 115,054	=	690,324
Steel plates	460	70 × 30,000 × 0.12	=	252,000
Radiation[b]	575	505 × 856 × 0.30	=	129,684
Total				1,072,008

[a]Binder oxidizes and liberates heat, which is neglected in this calculation.

[b]Average value of coefficient is less than 0.3 because oven is not up to 575 F. This is neglected. 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate.

Heat in 1 lb fuel oil	=		18,830 Btu
Heater Loss (10 per cent)	=	1883	
Duct Loss (5 per cent)	=	942	2,825 Btu
			16,005 Btu available to heat oven.

Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu (Fig. 8)

15 lb × 205	=	3,075 Btu sensible heat in products of perfect combustion.
		12,930 Btu to heat air X and Y (Fig. 11).

$$Y\,(S_{825} - S_{422}) + X\,(S_{825} - S_{70}) = 12930 \tag{4}$$

$Y = 2\,(X + 15)$ for 66.7 per cent recirculation.

where

S = heat content of air at temperature noted taken from Fig. 8

(Recirculation and exhaust contains water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air).

$S_{825} - S_{422} = 190 - 91 = 99$

$S_{825} - S_{70} = 190 - 8.6 = 181.4$

Substituting values of Y, H, etc. in Equation 4,

$(2\,X + 30)\,99 + 181.4\,X = 12,930$

$X = 26.3$ lb excess air.

$Y = 82.6$ lb recirculating air.

Total = 26.3 + 82.6 + 15 = 123.9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = 0.333 × 123.9 × ($S_{422} - S_{70}$) = 41.3 (91 − 8.6) = 3,400 Btu.

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel.

Fuel used in first hour 2,180,470 ÷ 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required. If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = 41.3 ($S_{675} - S_{70}$) = 41.3 (127 − 8.6) = 4880 Btu per pound fuel.

Heat available for heating material = 16,005 − 4880 = 11,125 Btu.

Fuel used in second hour 1,072,008 ÷ 11,125 = 96.5 lb oil = 14.3 gal.

Total oil used per load = 25.6 + 14.3 = 39.9 gal.

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8.5 F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air drier, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1.5 lb to a more usual figure of from 2.5 to 3 lb of steam per pound of water evaporated.

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PROBLEMS IN PRACTICE

1 ● What makes a commercial adiabatic drier differ from a theoretical one?

The word *adiabatic* means no heat lost to the outside and that the sensible heat lost by the air is equal to the latent heat of the water evaporated. In an actual drier, the solid containing the water, and the water itself must be heated to the temperature of evaporation, before evaporation can begin. Radiation losses from the drier enclosure is the other factor causing deviation from the theoretical adiabatic process.

2 ● What is a zone drier?

This term refers to a continuous drier where the drying medium is divided into two or more sections, in order to have better control of the temperature and humidity gradients through the drier, and often different velocities.

3 ● If a material enters a drier containing 70 per cent water and 30 per cent solids, and leaves the drier with 10 per cent water and 90 per cent solids, (a) What is the evaporation per pound of dried product? (b) What is the evaporation per pound of bone dry material?

a. $\frac{90}{30} - 1 = 2$ lb water per pound dried product.

b. Water entering $= \frac{70}{30} = 233$ per cent on bone dry basis.

Water leaving $= \frac{10}{90} = 11$ per cent on bone dry basis.

Water evaporated 222 per cent on bone dry basis.

Evaporation = 2.22 lb water per pound bone dry material.

4 ● What items must be included in a calculation of the drier heat requirements?

a. Heating water to be evaporated from the entering temperature to the temperature of evaporation.

b. Evaporating water to be removed.

c. Superheating evaporated water from the temperature of evaporation to the exit temperature of the air.

d. Heating material from entering to leaving temperatures.

e. Heating residual water from the entering to the leaving temperatures.

f. Heating conveyor or other supporting materials.

g. Radiation losses through the enclosure.

h. Sensible heat in the exit air.

5 ● The following conditions prevail in a drier; 250 lb water evaporated per hour. Air enters heater at 80 F dry-bulb and 65 F wet-bulb. Air exhausted from drier at 130 F dry-bulb and 100 F wet-bulb. Stock enters drier at 70 F. Heat required for warming stock and radiation losses are not considered. Fan is located ahead of heater. Find conditions of air entering and leaving drier, volume handled by fan, and temperature of air entering drier to supply the necessary heat, using Humidity Chart in Fig. 4.

Entering Air: Humidity, H = 0.01 lb water vapor per pound dry air.
Dew-point = 57 F
Per Cent Humidity, $\% H$ = 46

Leaving Air: Humidity, H = 0.0355 lb water vapor per pound dry air.
Per Cent Humidity, $\% H = 32$

Water pick up = 0.0355 − 0.01 = 0.0255 lb per pound bone dry air.

Bone dry air circulated per hour = 250 ÷ 0.0255 = 9800 lb.

Volume of air circulated at 80 F dry-bulb, and 46 per cent humidity 14.1 − 13.6 = 0.5 cu ft vapor (Fig. 4).

Volume = 13.6 + (0.46 × 0.5) = 13.87 cu ft = 1 lb dry air + vapor.

Volume handled by fan at 80 F $= \dfrac{9800 \times 13.87}{60} = 2260$ cfm.

Btu received by water = (130 − 70) × 1.0 = 60

Latent heat of steam at 130 F (Fig. 4) = 1019

Total = 1079 Btu per pound.

Heat used for evaporation per pound dry air = 1079 × 0.0255 = 27.43 Btu.

For entering air: Humidity, H = 0.01, Humid Heat = 0.2425 Btu per pound (Fig. 4)

$$(t_1 - t_2) \times S = \text{Btu for evaporation}$$
$$(t_1 - 130) \times 0.2425 = 27.43$$
$$t_1 = 247 \text{ F}$$

6 ● Given the following conditions, air 160 F dry-bulb, 49.6 per cent relative humidity (Φ), 29.92 in. Hg, barometric pressure, find the per cent H, absolute humidity.

For 160 F, p_s = 9.65 in. Hg (From Table 6, Chapter 1)

$$p = \Phi p_s = 0.496 \times 9.65 = 4.78$$

$$\frac{29.92 - 9.65}{29.92 - 4.78} \times 0.496 = 0.40 \text{ or 40 per cent absolute humidity.}$$

Chapter 36

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases with mechanical forces, finds extensive application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are the wind and the difference in temperature of the air inside and outside the building. The arrangement and control of ventilating openings should be such that the two forces act cooperatively and not in opposition.

Wind Forces

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings and trees.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \tag{1}$$

where

Q = air flow in cubic feet per minute.
A = free area of inlet (or outlet) openings in square feet.
V = wind velocity in feet per minute,
= miles per hour $\times$ 88.
E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle.)

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E.

The accuracy of the results obtained by the use of Formula 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edge orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less, and if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following four places:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. In a monitor on the side opposite from the wind.
4. In roof ventilators or stacks exposed to the full force of the wind[1].

Forces Due to Stack Effect [2]

The stack effect produced within a building is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\,\sqrt{H\,(t - t_0)} \tag{2}$$

where

Q = air flow in cubic feet per minute.
A = free area of inlets or outlets (assumed equal) in square feet.
H = height from inlets to outlets, in feet.
t = *average* temperature of indoor air in height H, in degrees Fahrenheit.
t_0 = temperature of outdoor air, in degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

The height between inlets and outlets should be the maximum which the building construction will allow.

In some cases the necessary air flow will be known from the requirements of the building occupancy, and the area necessary for certain assumed temperature differences may be calculated. Or the areas may be fixed by the building construction, and the maximum air flow for various differences between indoor and outdoor temperatures may be calculated. In any case, the conditions which give the minimum air flow are those which control the design, as the system must have ample capacity even under the most unfavorable conditions which are those of mild or warm weather.

TYPES OF OPENINGS

The engineering problems of a natural ventilation system consist of the *design, location, and control of ventilating openings* to best utilize the

[1]Airation of Industrial Buildings, by W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 159).
[2]Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926. p. 59).
Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 605).

natural ventilation forces, in accordance with the requirements of building occupancy. The types of openings may be classified as:

1. Windows, doors, monitor openings, and skylights.
2. Roof ventilators.
3. Stacks connecting to registers.
4. Specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in

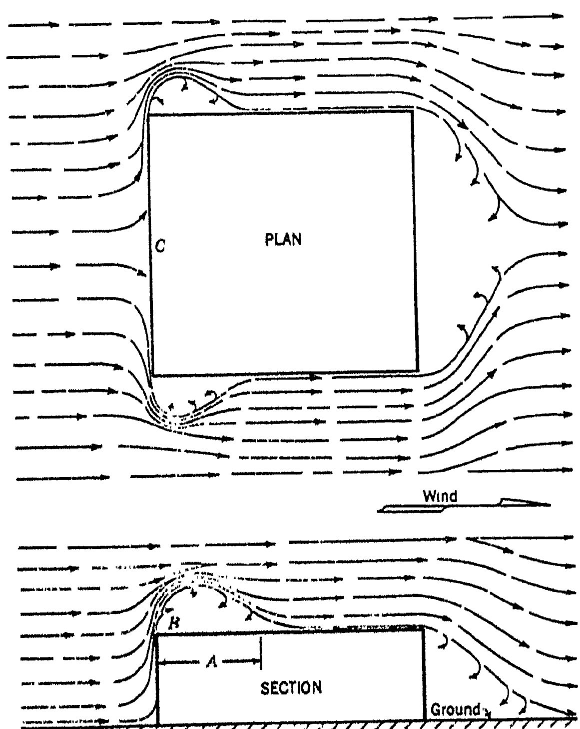


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top or bottom. Whatever the form and type of window used, the amount of clear area that can be made available is the factor of greatest importance in ventilation.

All types of sash (double-hung, top, center or bottom horizontal pivoted, or vertical pivoted) have about the same air flow capacity for the same clear area. Air leakage through *closed* windows is important during high winds (Chapter 6).

The proper distribution of air in occupied spaces is an element almost as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Door openings are seldom included in the ventilation calculations, though they may be of great value for extreme summer conditions, and should be considered in this connection as well as in garage design.

Skylight and monitor openings are of importance as these and the roof ventilators are outlets, while the lower windows are usually inlets on the windward side and outlets on the leeward side. In general the areas of inlets and outlets should be about equal. It is important to make a check on this ratio in any installation, as any great excess of area of one set of openings over another means waste opening area. The operating devices used for sash, monitors, skylights and roof ventilators should be well selected as poor operating devices may defeat the entire design.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet, which is sensitive to wind action for producing additional flow capacity, and at the same time is subject to manual or automatic control by suitable dampers. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it offers to air flow, (3) the area and location of openings provided for air inflow at a lower level, and (4) the ability of the ventilator head to utilize the kinetic energy of the wind for inducing flow by centrifugal or ejector action. Frequently one or more of these capacity factors is overlooked in a ventilator installation.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1.) This does not mean that no ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little importance in these locations, and hence their size should be increased proportionally.

Ventilator resistance depends on (1) type of inlet, (2) area of openings and passages, and (3) number of turns or changes of direction of the air flow. The inlet grille, if any, should have ample free area, and the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of a square-entrance orifice (flow coefficient 0.60). In other words, the grilles should be oversize as compared with the ventilator, and they should be connected by tapering collars. If the ventilator head construction produces changes in the direction of air flow, the area of the flow passages should be increased accordingly.

Air inlet openings at lower levels in the building are of course necessary for the economical use of ventilator capacity. The inlet openings should be at least equal to, and preferably twice as great as the combined throat areas of all roof ventilators. The air discharged by a roof ventilator

depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 11. These may be classified as *stationary*, Figs. 2 to 6, *pivoted* or *oscillating*, Figs. 7 to 9, or *rotating*, Figs. 10 and 11. When selecting roof ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

It should be kept in mind that a suitable combination of roof ventilators with mechanical ventilation frequently offers the best solution of a ventilating problem. The natural ventilation units may be used to supplement power driven supply fans, and under favorable weather conditions it may be possible to shut down the power driven units. Where low operating costs are very important, such a combination has great advantages. Roof ventilators with built-in electric fans are attracting increased attention because they combine the advantages of low installation and operating cost with those of continuous service.

Controls

In connection with any combination between natural and fan ventilation, the controls are of importance. Both the fans and the ventilator dampers may be controlled by some combination of three methods: (1) hand operation, (2) thermostat operation, and (3) control by wind velocity. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called *gravity* action). While their openings projecting above the roof are not provided with any special construction for developing suction by the action of the wind, the plain vertical opening is also effective in this respect. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

Stacks are applicable particularly in the case of schools, apartments, residences and small office buildings. Partitions interfere with general air circulation, and some type of outlet from each room is necessary. If the building is not too tall, and the requirements of occupancy are moderate, a system of stacks with registers in each room may be more economical than a system of mechanical ventilation employing fans. In making the comparison, however, the building space occupied by the stacks should be considered.

With little or no wind, chimney effect or temperature difference will produce outflow through the stacks and an equal inflow through windows

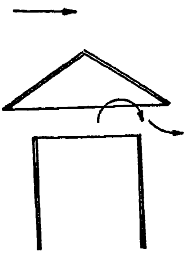


FIG. 2

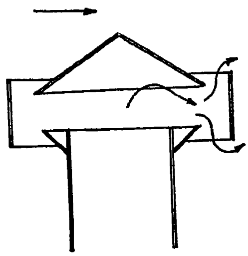


FIG. 3

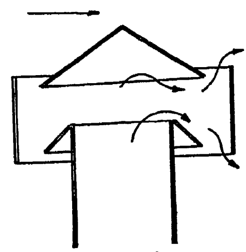


FIG. 4

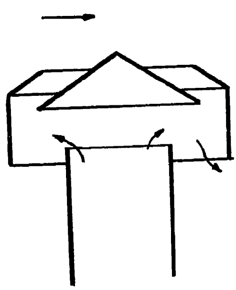


FIG. 5

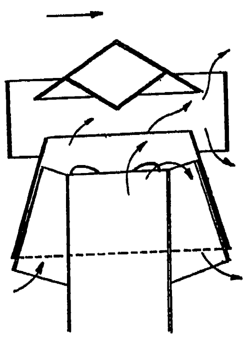


FIG. 6

FIVE COMMON TYPES OF STATIONARY VENTILATORS

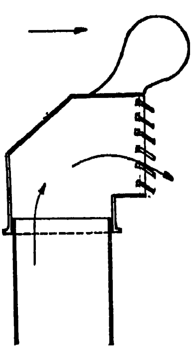


FIG. 7

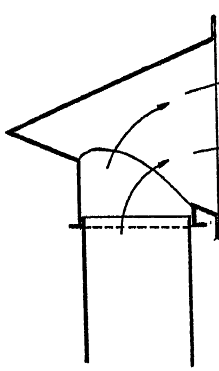


FIG. 8

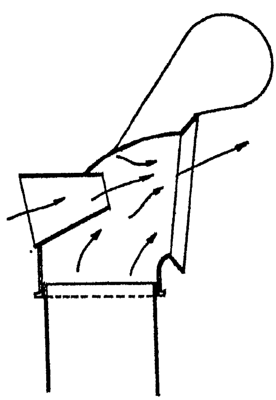


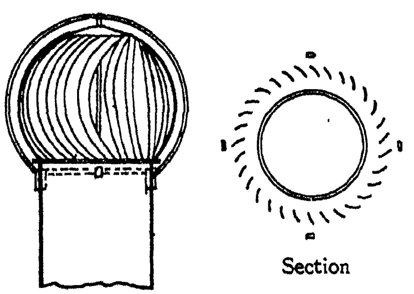
FIG. 9

THREE TYPICAL OSCILLATING VENTILATORS

in all sides of the building. With wind, the inductive force at the top of ventilating shafts is more powerful than that on the leeward side of the building, so that air is drawn in through leeward openings by a combination of the forces of wind and temperature difference. On the windward side, the direct forcing pressure of the wind is of course added to the temperature difference effect. Thus forces are available for causing inflow at practically every window of such a building. Adequacy of stack size must, of course, be provided.

PRINCIPLES OF AIR FLOW CONTROL

The air flow through a ventilation opening depends on the two factors already discussed, namely, (1) the natural forces available, (2) the openings available, and the resistance to flow offered by these openings. The design problem includes, of course, a determination of the *desired air*



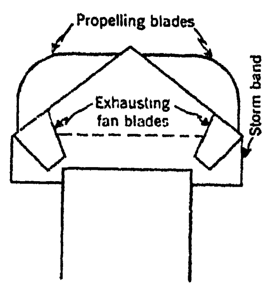


FIG. 10. ROTATING VENTILATORS FIG. 11.

quantity and distribution in order that the openings may be properly placed.

The purpose of ventilation is to carry off either excess heat or air impurities, and the desired air quantities depend upon the amount of heat or of impurities present. The amount of heat can be determined, in the case of forge shops for example, from the amount of fuel burned, which in turn is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some instances, not all of the heat may be dissipated to the air, but a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$H = \frac{c\,60\,Q\,(t - t_0)}{V} \tag{3}$$

where

c = 0.24 = specific heat of air.

V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 1.)

H = heat to be carried off, in Btu per hour.
Q = air flow in cubic feet per minute.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.

For disposing of air impurities, the required air flow must be such that the outside air will dilute the impurities to a degree that they are no longer objectionable. For human occupancy, such as in auditoriums and classrooms, 10 cfm per person is usually taken as the minimum of outside air necessary for ventilation (see Chapter 3). For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Air *quantity* and *quality* are not the only requirements. For human occupancy, air *distribution* is important. In ventilation the air distribution is almost entirely a matter of the number, the design, and the location of inlets and outlets. In locating openings, special precautions should be taken against the formation of dead air spaces or *pockets* within the zone of occupancy (see Chapter 28).

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity[3], this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations[4] show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicted with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Temperature differences are 10 F in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per hour is:

$$H = 15 \times 7.75 \times 18{,}000 = 2{,}092{,}500 \text{ Btu.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{c\,60\,(t - t_o)} = \frac{13.5 \times 2{,}092{,}500}{0.24 \times 60 \times 10} = 196{,}172 \text{ cfm.}$$

[3]Loc. Cit. Notes 1 and 2.

[4]This is true for *turbulent* flow only. It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings.

This is equal to 19.6 air changes per hour. The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening. Actually, the latter difference is larger and so the value of 19.6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2:

$$A = \frac{Q}{9.4\sqrt{H(t - t_o)}} = \frac{196{,}172}{9.4\sqrt{30 \times 10}} = 1{,}205 \text{ sq ft.}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0.50 \times 1{,}205 \times 5 \times 88 = 265{,}100 \text{ cfm.}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified. Apparently 1205 sq ft are a very generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example, it must be apparent that while formulas may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulas.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone of occupancy.

2. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.

4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

5. In an industrial building where furnaces, that give off heat and fumes, are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

n case it is impossible to locate furnaces in the windward end, that part of the g in which they are to be located should be built higher than the rest, so that nd, in splashing therefrom will create a suction. The additional height also es the effect of temperature difference to cooperate with the wind.

n the use of monitors, windows on the windward side should usually be kept since, if they are open, the inflow tendency of the wind counteracts the outflow cy of temperature difference. Openings on the leeward side of the monitor result ›eration of wind and temperature difference.

n order that the force of temperature difference may operate to maximum advan- he vertical distance between inlet and outlet openings should be as great as e. Openings in the vicinity of the neutral zone are less effective for ventilation.

n order that temperature difference may produce a motive force, there must be l distance between openings. That is, if there are a number of openings available ıilding, but all are at the same level, there will be no motive head produced by ature difference, no matter how great that difference might be.

In the design of window ventilated buildings, where the direction of the wind is constant and dependable, the orientation of the building together with amount ouping of ventilation openings can be readily arranged to take full advantage of ce of the wind. On the other hand, where the direction of the wind is quite e, it may be stated as a general principle that windows should be arranged in lls and monitors so that there will be approximately equal area on all sides. 10 matter what the wind's direction, there will always be some openings directly d to the pressure force of the wind, and others opposed to a suction force, and e movement through the building will be assured.

The intensity of suction or the vacuum produced by the jump of the wind is t just back of the building face. The area of suction does not vary with the wind y, but the flow due to suction is directly proportional to wind velocity.

Openings much larger than the calculated areas are sometimes desirable, especially hanges in occupancy are possible, or to provide for extremely hot days. In the case, free openings should be located at the level of occupancy for psychological ı.

Special consideration should be given to the possibility of sidewall or monitor ‹s being closed on account of weather conditions. Such possibilities favor roof tors and specially designed stormproof inlets.

MEASUREMENT OF NATURAL AIR FLOW

: determination of the performance of any ventilating system es measurements which are not easy to make. The difficulties are sed in the case of natural ventilation, since the motive forces and r velocities are very small. The measurements necessary for giving *pacity* of a system are (1) velocity of the wind, (2) velocity of the rough inlet and outlet openings, (3) outdoor air temperature, and erage indoor air temperature.

ısuring Wind Velocity. The cup-type of anemometer as used for ıer Bureau observations is sufficiently accurate for this measure- Some more accurate instruments as well as direct-reading types een developed for airport service, but for ventilation work it is the ge wind velocity over a long period which determines the capacity of stem. Hence the use of the Weather Bureau instrument, with an ation period of one hour or more, is satisfactory. If observations ıd direction are required, these should be taken by observing a ve weather vane at frequent intervals (about every 5 minutes) ; the same period.

ıcity of Air Through Openings. The vane type anemometer is the ›ractical instrument for this measurement.

Use a small (4 in.) low-speed anemometer, and correct all readings according to a recent calibration. Mount the anemometer in a strap iron clamp with a long handle for convenience. Divide each opening into 5 in. squares (by string or wire) and hold the anemometer in the center of each square for a definite period of from 15 to 30 seconds. Record the result of the traverse as soon as completed and start another one immediately. A series of traverses over a period of one hour, or the full period covered by the wind velocity observations with a fairly steady wind, may be considered a satisfactory test for that wind velocity. It is preferable to have an anemometer observer at each opening. If the opening is covered by a grille or register, use the proper correction factors (see Chapter 44).

Outdoor Temperature. It is easy to make an error of 1 to 5 deg in observing the outdoor air temperature. An accurate thermometer, calibrated in 1 deg divisions should be used. The thermometer should be mounted in the shade at about mid-height of the building and not too near the building wall or adjacent to an air outlet. The heat from a wall or roof which has been exposed to the sun is easily transmitted to a thermometer, with resulting high readings.

Average Indoor Temperature. It is important to note that the capacity of an opening (such as roof ventilator) does *not* depend on the difference in the temperatures measured adjacent to the opening. It depends rather on the difference between the *average* temperature of the column of air inside the building and that outside. Indoor temperatures should therefore be observed at various heights to secure a good average.

DAIRY BARN VENTILATION [5]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be

[5] Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 181).

Cow Barn Ventilation, by Alfred L. Offner (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 149).

For additional information on this subject refer to *Technical Bulletin*, U. S. Department of Agriculture (1930), by M. A. R. Kelley.

supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cfh of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cu ft of barn space, and 0.197 to 0.305 Btu per hour per sq ft of barn exposure.

GARAGE VENTILATION [6]

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929 and revised in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the

[6]Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355), (A.S.H.V.E. Reprint, January, 1935).

Airation Study of Garages by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 233).

Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 511).

Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 439).

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 424).

Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 263).

University of Kansas, Lawrence, Kans., in cooperation with the A.S.H.V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed below:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

PROBLEMS IN PRACTICE

1 ● What factors may make the adoption of a system of ventilation depending upon wind movement inadvisable in new construction?

a. Variation in direction of wind.
b. Variation in wind velocity.
c. Inability to clean incoming air.
d. Inability to control location, size and shape of buildings on adjacent property.
e. Unsatisfactory warming of incoming air during cold weather.

2 ● a. What factors are important in the location and control of ventilating openings?

b. What types of ventilating openings are best suited to a proper distribution of the air supplied?

a. The proper distribution of air as required by the occupants, and the best utilization of natural ventilating forces. The general rules referred to in this chapter apply particularly to these factors.

b. Windows with swinging sash and openings with deflectors may be used to direct air to the points desired.

3 ● a. What is the best location for ventilating openings?

b. How are the sizes of ventilating openings determined for proper air supply?

a. Inlet openings should be low and facing the prevailing winds where possible. Outlet openings should be high and on the side opposite the prevailing winds.
b. For simple openings use Formula 1:

$$Q = EAV$$

and for stacks use Formula 2:

$$Q = 9.4\,A \sqrt{H\,(t - t_o)}$$

The use of these formulae is illustrated in Example 1 of the text of this chapter. Inlet and outlet areas should be approximately the same for best results.

4 ● a. What are the advantages of roof ventilators?

b. How are proper sizes determined for roof ventilators?

a. Roof ventilators offer the best utilization of the inductive force of the wind, and they may be very economically fitted with built-in fans to supply the necessary circulation when the force of the wind is not sufficient.

b. Because of the many factors affecting the flow through roof ventilators no accurate formula can be given. It is usual practice to make the combined throat area of all roof ventilators between one-half area and full area of the air inlets as determined by Formula 1.

5 ● What methods of control are used in ventilating systems?

Hand control, control by a thermostat located in the ventilated space or in the ventilator, or wind velocity control designed to keep the air discharge constant regardless of wind velocity.

6 ● How is the quantity of air required for a building determined?

Sufficient air must be supplied to carry away the heat and impurities generated within a building. The temperature rise and concentration of impurities in the exhaust air must be held within specified limits. (See Example 1 in the text of this chapter).

7 ● What measurements are necessary to determine the capacity of a ventilating system?

Wind velocity and air velocities through openings, determined by suitable anemometers; outdoor air temperatures, measured by a shaded thermometer not near objects heated by the sun or near exhaust air openings; indoor air temperatures, measured at various heights to secure a good average.

8 ● How much air must be supplied for dissipating the heat generated in a dairy barn housing 100 cows if the outside temperature is 20 F and the inside temperature is to be maintained at 45 F?

The total heat generated is 100 × 3000 = 300,000 Btu per hour. Then from Formula 3,

$$Q = \frac{VH}{c\,60\,(t - t_0)}$$
$$= \frac{13.5 \times 300{,}000}{0.24 \times 60\,(45 - 20)}$$
$$= 11{,}250 \text{ cfm.}$$

This amount of air should also keep down humidity and odors.

9 ● a. What precaution is necessary in the ventilation of garages using natural ventilation?

b. How much window area is required for a garage with 50 x 100 sq ft floor area if natural ventilation is used?

a. The carbon monoxide content of the air should be kept below 1 part in 10,000 and windows should be kept open at all times.

b. The window area should aggregate 5 per cent of the floor area.

$$0.05 \times 50 \times 100 = 250 \text{ sq ft of window area.}$$

This area should be evenly distributed along two sides of the building.

Chapter 37

AUTOMATIC CONTROL

Purpose of Automatic Control, Definitions of Control Units and Terms, Types of Control, Central Fan Systems, Unit Systems, Control of Automatic Fuel Appliances, Residential Control Systems, Control of Refrigeration Equipment, Industrial Processes

THIS chapter is prepared with the purpose of acquainting the engineer with the principles underlying the use of automatic control, the general types and varieties of control equipment available and their application.

Automatic control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. A properly designed and complete control system has the ability to interlock and coordinate the various functions of heating, ventilating and air conditioning in a manner impossible to accomplish with manual regulation.

Automatic control is an integral and essential part of a heating, ventilating or air conditioning installation and cannot be regarded as an accessory. In order to insure satisfactory results, the control should be designed with and incorporated in the heating, ventilating or air conditioning system. The control equipment should be given careful consideration in the planning of any installation in order that the entire system may operate together with satisfactory results.

In order that proper selection and application of controlling devices may be made it is important that a broad understanding exist as to the types of control available and their principles of operation. Improper selection and application of control equipment will result in unsatisfactory and inefficient operation. Specific control devices and systems are described in the *Catalog Data Section.*

PURPOSE OF AUTOMATIC CONTROL

Automatic control is normally applied to heating, ventilating or air conditioning systems:

1. To insure the maintenance of certain desired or required conditions of temperature, pressure, humidity, air motion or air distribution.

2. To serve a safety function, limiting pressures or temperatures within predetermined points, or preventing the operation of mechanical equipment unless it may function without hazard.

3. To produce economical results and thereby insure operation of the system at a minimum of expense.

DEFINITIONS OF CONTROL UNITS AND TERMS

Controlling devices and terms commonly used in the automatic control of heating, ventilating and air conditioning systems are:

Thermostats: Thermostats are defined as temperature sensitive devices reacting to temperature changes. There are four major types of thermostats.

A Room Thermostat is normally installed on the wall of the room whose temperature it is to control, and in reacting to rising or falling temperatures, the thermostat causes the operation of heating or cooling equipment so that desired temperatures will be maintained.

The temperature sensitive element will usually consist of a bi-metal strip or coil, or a vapor-filled bellows as illustrated in Fig. 1.

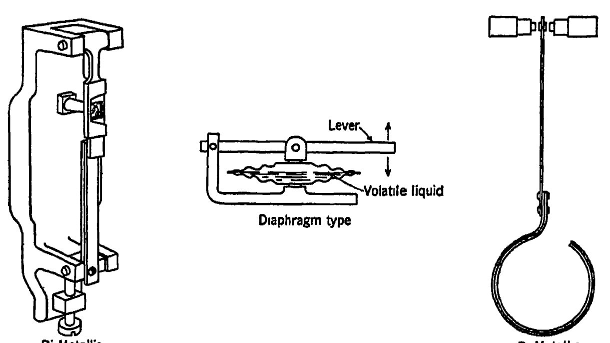


FIG. 1. TYPICAL THERMOSTATIC ELEMENTS

Immersion Thermostats are used for controlling liquid temperatures. The sensitive element will normally be encased in a protective well which is inserted in the liquid, the temperature of which is being controlled.

The temperature sensitive element will usually consist of a bi-metal coil, thermal expansion rod, or a vapor-filled system. If the latter is used the temperature sensitive bulb may be connected to the case of the instrument by either a flexible or rigid tube.

Insertion Thermostats are similar to immersion thermostats except that they are for use in controlling the temperature of a gas such as air. The sensitive element will often be encased in a protective well which prevents mechanical damage but which permits the gas to come in direct contact with the element.

Surface Thermostats include those devices which measure surface temperatures. These surface temperatures will often be an indirect measure of the temperature of a gas or fluid as in the case of a pipe within which water is flowing. The sensitive element will usually be placed in direct contact with the surface of the object whose temperature it is to measure and may consist of a bi-metal spiral or vapor-filled bellows.

Humidity Controls: Humidity controls are defined as automatic devices reacting to changes in relative humidity. Within this group, the devices which operate in controlling humidity supplying equipment are regulating devices and when operating only to prevent relative humidity from exceeding a predetermined maximum are a form of limit control.

The humidity sensitive element may consist of hair, paper, wood, skin or any other material which changes its dimensions with changes in humidity.

Controls are available provided with both temperature and humidity sensitive elements, which operate to maintain definite relations between dry-bulb temperature and relative humidity.

Pressure Controllers: Pressure controllers are defined as devices reacting to pressure and pressure changes. Examples of such devices are the pressure controls governing the operation of refrigeration equipment from either head or suction pressure, devices reacting to steam or water pressure or the pressure of air in the distribution systems.

Damper Motors: Damper motors are defined as specialized power units, the purpose of which is to position outdoor air, face, by-pass or distribution dampers, regulating the flow of air through the system. Connected by suitable linkages, these damper motors react at the command of thermostats, humidity controllers and pressure controllers to adjust the air flow to the needs of the system.

Control Valves: Control valves are defined as steam valves, water valves or air valves which may be adjusted at the command of controllers to regulate the flow of the medium passing through them to the needs of the system. Such control valves are usually constructed with an electric or pneumatic power unit connected to the valve stem so that the movement of the power unit will react to position the valve as conditions demand.

Self-contained valves are also included under this classification. Their application is principally limited to the regulation of the steam supply to individual radiators in two-pipe low pressure steam heating systems, and the temperature of hot water supply tanks.

Solenoid Valves: Solenoid valves are, as their name implies, valves actuated by the magnetic effect of an electric solenoid built within them. While normally these valves are opened when the solenoid is energized, they are sometimes built in a reverse acting manner and closed when energized. In heating, ventilating and air conditioning systems, they are normally adapted to the control of oil or gas burners as fuel valves, as water valves on humidifiers, or as refrigerant valves in refrigeration systems.

Relays: A relay is defined as a unit installed between a controller and the device under control, for purposes of amplifying the capacity of the controller or performing an auxiliary control function. For example, a thermostat, in order to preserve its sensitivity may be constructed so that it is not capable of handling the power required of a motor. A relay is, therefore, installed between the two. The thermostat actuates the relay and the relay, in turn, actuates the motor. Motor driven switching devices are also often used as relays.

TYPES OF AUTOMATIC CONTROL

Operating Media or Source of Power Supply

Automatic control systems may be classified in three broad groups based upon their primary operating media or source of power, as follows:

1. *Electric Control Systems.* In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays or solenoids. The individual units of this type of system are interconnected by line voltage or low voltage wiring, and this wiring serves to complete the circuits carrying the commands of the controllers to the controlled valves or damper motors.

2. *Pneumatic Control Systems.* In the pneumatic control systems, the primary source of operation is obtained through a medium of compressed air, the pressure of which is varied by the controlling devices. In these systems one or more centrally located air compressors furnish a supply of compressed air which is distributed in special piping to the various controlling and controlled devices. By means of leak ports or orifices, the pressure of the air is varied in the branch lines and the changing pressures are utilized in air operated damper motors or valves to obtain the movement necessary to the operation of valves and dampers.

3. *Self-Contained Control Systems.* Self-contained control systems have, in general, been restricted to such operations as could be effectively handled by a power unit with integrally mounted or direct-connected controller. Such applications consist of valves utilized to admit steam or other media into coils to regulate the temperature of tanks or to regulate the admission of steam into heating coils or radiators as determined by the controller element.

Motion of Controlled Equipment

Automatic control equipment can also be classified into two general types with respect to the characteristics of the motion imparted by the controls to the controlled equipment, such as two position or positive acting control and modulating or graduated action control.

In any control system it is necessary to choose the type of equipment whose characteristics permit the type of control operation desired and in many cases both types of control are used in the same system to best meet various requirements.

1. *Two Position or Positive-acting Control.* This type of control operates positively between two positions such as *on and off* or *open and closed* with no intermediate positions or degrees of motion between the two extremes of operation. A simple thermostat which starts and stops an oil burner or a unit heater motor is an example of this type. As applied to a valve or a damper, the action of the controlling device would serve to fully open or fully close the valve or damper.

In some applications of this type of control, artificial heat is applied to the sensitive element of the room thermostat at the same time that heat is being added to the space under the control of the thermostat in order to

increase its sensitivity. This usually results in more accurate control and more frequent operation of the heat source.

2. *Modulating or Graduated-acting Control.* This type of control causes motion in the controlled device in proportion to motion caused in the controller by fractional degree variations in the medium to which the controller is responsive. After a fractional change has been measured at the controller and has effected a new position of the valve or damper in proportion to the amount of such change, the system stands by awaiting further change at the controller before any additional motion occurs. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of conditions as measured. With this type of control, the damper or control valve may be operated in intermediate positions between its extreme limits in order to properly modulate or proportion the flow of air, steam or water, reacting with changes of conditions at the controller. Various modifications of this type of control are available, designed to meet special requirements and conditions, all based on operation of the controlled equipment in intermediate positions.

This type of control motion cannot be used on valves of one-pipe steam systems as the partial opening of the valves will not permit the condensate to escape against the flow of incoming steam. This type of control should not be used to control the flow of steam to a heater coil of a fan system which is in the direct path of untempered outdoor air at temperatures below freezing, because of the possibilities of freezing condensate in the bottom of the coil.

Division of Space under Control

Control systems vary considerably with the type and size of the building, occupancy of the building, and with the heating or cooling system, humidity supplying equipment and ventilating means available for control. In the following paragraphs the general requirements of various phases of these different buildings will be discussed.

1. *Individual Room Control.* The most accurate and flexible form of control for any structure is that calling for the regulation of each individual room by control equipment reacting to conditions in that room only. Such control necessitates a thermostat in each room, located to properly measure the conditions of the room, controlling the radiator, unit heater, unit ventilator or other heating source supplying heat to that room only in which the thermostat is located. This arrangement permits the maintenance of any desired conditions in any room, entirely independent of any other room. In the case of large rooms, where one thermostat location will not serve to properly measure the conditions throughout the room, and where two or more sources of heat supply are provided in the room, additional thermostats may be used, each controlling its respective section of the heating source. This form of control, due primarily to the number of control devices required over the entire building, normally is the most expensive type of control system. However, where maximum flexibility and the most accurate control is desired, individual room control can be depended on to furnish the desired results.

2. *Single Thermostat Control.* Probably more widely used than any other form of control is the type of automatic system regulated entirely

from a single room thermostat. The wide use of this particular means of control is primarily due to the fact that it is the form of regulation best adapted to residences and small buildings, which far out-number the larger structures. In larger buildings, this form of control has definite shortcomings. In the small buildings and average size residences it is possible to select a location and install a thermostat of suitable characteristics which, in controlling from the surrounding air temperature, will hold the temperature of the entire building within entirely satisfactory limits. It must be recognized that the thermostat reacts to and controls from the temperatures to which it is subjected and that, therefore, the position selected for the thermostat must be representative of general conditions throughout the structure. It must further be recognized that if certain areas or rooms of a structure are not properly balanced as regards heating or cooling capacity and distribution, the control as dictated by the thermostat will not produce satisfactory results in these unbalanced areas.

3. *Zone Control.* As the size of buildings increases, it becomes increasingly difficult to provide proper regulation for the entire structure from a single thermostat control. In such instances, where the advantages of individual room control are not obtainable by reason of its cost, an intermediate form of control system is available, commonly described as *zone control.* In this form of control system a building is divided into areas or zones such that the general requirements and the general conditions through the areas are relatively constant as to exposure and occupancy, and then each zone is provided with control equipment which functions to regulate the conditions in that particular zone. As in the case of individual room control, each zone may be regulated to its own needs which may vary from the needs of other zones within the same structure.

Variations of the usual zone control methods by the use of recently developed special devices have been quite successful in obtaining greater economy from heating systems. Frequently these use an outside thermostat or group of thermostats which adjust the operation of the controls to conform to variations in weather conditions.

CENTRAL FAN SYSTEMS

A central fan system includes any conditioning system by which either outdoor air, return air, or combinations of outdoor and return air, are conditioned at a central point and then distributed through duct work to the various sections of the space being conditioned.

Heating Cycle

Central fan ventilating systems may be sub-divided, first into split systems, by which air is supplied for ventilating purposes only and heat is supplied in winter from another source such as direct radiation; and second, into combined systems, in which the functions of ventilation and heating are both performed by the central fan system.

A control system for a central fan ventilating system using all outdoor air and discharging air at a predetermined temperature is illustrated in Fig. 2. Thermostat T_1 located in the outdoor air intake is set just above freezing, and controls valve V_1 on the first heating coil. This valve must

be completely open or completely closed to avoid danger of freezing. The by-pass damper around the heaters and the other two valves V_2 and V_3 are controlled by thermostat T_2 located in the discharge duct from the fan. If the temperature of the discharge air increases, through the action of T_2 the damper is moved automatically to admit more cold air. Should this not reduce the temperature sufficiently, the valves V_2 and V_3 on the heating coils will be closed gradually and in sequence until the correct temperature is reached. The control of the damper and valves V_2 and V_3 must be gradual or there will be a wide fluctuation in temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the room and in order to maintain controlled room temperatures it is necessary to control the radiators independently of the ventilation control.

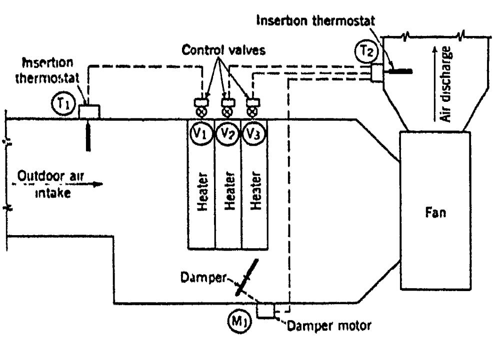


FIG. 2. CONTROL OF A SPLIT SYSTEM OF VENTILATION

In some installations, such as theatres and auditoriums, it is difficult to install sufficient direct heating surface to offset the heat losses from the room. There are also installations where a short heating-up period is allowed before occupancy of the room, and in these cases it is necessary to use the entire heating capacity of the ventilating system for this purpose. An additional thermostat may be installed in the room which will take the control away from the fan discharge thermostat (T_2 in Fig. 2) and utilize the full heating capacity when the room is below normal temperature.

In central fan systems, air washers are often used and in such cases, due to the effect of temperatures on humidity, additional control is required. An arrangement with control of the second tempering heating coil from the air washer temperature and with the usual control of the first tempering heating coil from the outside temperature is shown in Fig. 3. This permits the air to be kept cool while passing through the washer so that too much moisture will not be absorbed. Control of the reheating units and by-pass damper by an insertion thermostat in the fan discharge, and the application of a pilot thermostat to a system of this sort is illustrated in Fig. 3.

Where a number of rooms are to be heated and ventilated through one central fan system it is customary to provide tempering heating units, automatically controlled to provide a minimum temperature for ventilation only and additional heating units to supply the heating requirements. These reheating units may be located in the various branch ducts to the different rooms, each under control of its individual room thermostat, or individual ducts may be run to the various rooms from the central unit. In this case reheater coils are provided to maintain a predetermined temperature in a warm air chamber. Each room duct is connected to this warm air chamber and to the tempered air supply, and through the action of a room thermostat on a gradual-acting double-mixing damper the proper proportions of warm and tempered air are secured to maintain desired conditions in the room.

In all types of central fan systems, the outdoor air damper is usually opened and closed by a damper motor controlled from a manual switch

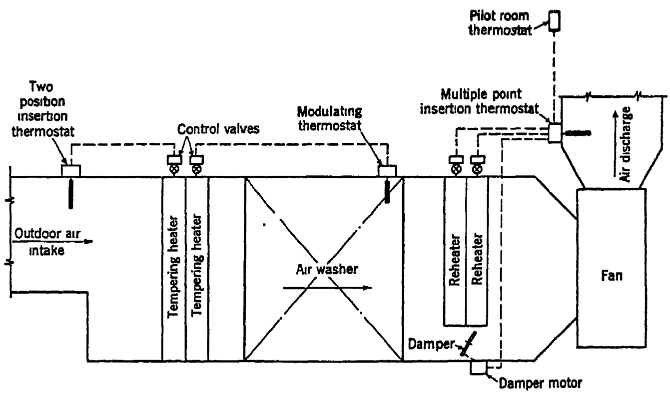


FIG. 3. CONTROL OF VENTILATING SYSTEM WITH AIR WASHER USING PILOT THERMOSTAT

or by a relay in the fan motor circuit, so arranged that when the fan motor is started, the relay causes the damper motor to open the outdoor air damper.

Recirculating and vent dampers may also be opened and closed by means of damper motors controlled from remote locations. Generally these damper motors are positive acting and are either completely open or closed. However, in some cases, where part outdoor air and part recirculated air is desired, it is advantageous to control the dampers so that definite proportions of damper opening area exist. In some installations the control of outdoor air and recirculating dampers is under the command of a thermostat at the intake to the conditioner, in which case the proportions of outdoor and recirculated air are fixed by the resultant temperature of their mixture. This arrangement tends to reduce the amount of outdoor air used as the outside temperature is lowered.

The operation of a central fan system during the heating cycle often results in unfavorably low relative humidity and the provision and control of humidity becomes an important factor of the system. If water spray humidification is used, control may be effected by a humidity con-

troller actuating a control valve in the water supply to the sprays. If steam humidification is used, either of the steam jet type or of the steam heated evaporating type, the flow of steam may be controlled from the humidity controller in the ventilated space. Where an air washer is used, approximate control of humidity may be obtained by maintaining the air temperature in the air washer at a predetermined desired dew-point temperature.

For example, the dew-point temperature at 70 F and 40 per cent relative humidity is 45 F. Therefore, if the air temperature is maintained at 45 F as it leaves an air washer (assuming it is fully saturated) and then is heated to 70 F, it will have a relative humidity of 40 per cent. If it is desired to maintain these conditions in a given space, the air temperature can be raised to any necessary point, say 120 F (at which the relative humidity will be only 9 per cent). When the heat in the air has been dissipated, the space temperature being maintained at 70 F, the relative humidity will be 40 per cent.

Whenever moisture is being added to the air during the heating cycle by the use of a spray or any other means, a considerable amount of care must be used in order to prevent frost from collecting on the windows due to the air being reduced below its dew-point at the inside surface of the windows.

Cooling Cycle

Central fan cooling systems are divided into two general groups based upon the methods employed to control the temperature and humidity of the treated space. Cooling normally involves the removal of moisture from the air, and to accomplish this end the temperature of the air must be lowered below the dew-point. The air at this low temperature must then be treated or introduced into the room in such manner as to avoid uncomfortable cold drafts.

In the first group the air is supplied from the conditioner after being cooled and dehumidified to a fixed temperature and humidity and then before entering the treated space is reheated. This is accomplished either by passing the air through coils heated with steam, hot water, or other heating medium, or the air from the conditioner is mixed with recirculated air before entering the conditioned space.

In the second group are those systems which use the treated space as a mixing chamber, the air being supplied to it at the temperature and humidity leaving the conditioner and depending upon diffusion in the conditioned space to give ultimately the correct conditions. In these systems the temperature and humidity of the treated space are measured and govern, through control of the cooling means, the temperature and the humidity of the air leaving the conditioner.

In Fig. 4 is represented one of the most simple central fan types of cooling system. Thermostat *T* measures the temperature within the treated space and operates to start and stop the refrigeration compressor or to control the supply of refrigerant to the cooling unit as required to maintain a fixed temperature in the space.

There are three general methods for the control of relative humidity in central fan cooling systems, which are:

1. By provision for limiting the relative humidity in addition to the temperature at a definite point. When this method is used, either temperature or humidity may demand operation of the cooling source regardless of whether or not the other factor has been exceeded. The use of a high limit humidity control in this manner is desirable during conditions of high relative humidity but its operation may cause excessive cooling unless some method of reheating is employed.

2. By the maintenance of a fixed effective temperature. By this method, a definite relation is maintained between temperature and humidity, and sensible cooling is done whenever possible instead of the removal of latent heat in the form of moisture.

3. By the maintenance of a fixed dew-point in the air discharge. This method usually provides for the control of relative humidity within the space being conditioned between reasonable limits, but does not take into consideration any change in the latent heat load, as compared to the sensible heat load.

The necessity for varying inside temperature conditions in accordance with changes in outdoor conditions on many types of installations is important. A control system is shown in Fig. 5 where the temperature of the treated space is adjusted according to the outdoor temperature.

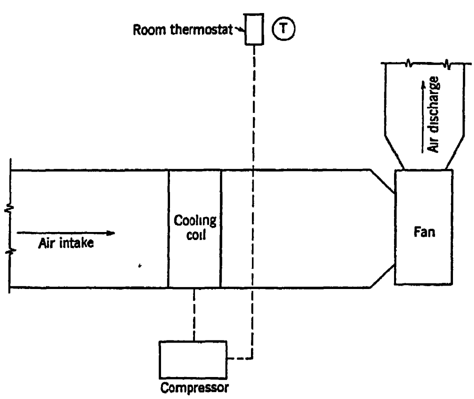


FIG. 4. DIAGRAM OF SIMPLE COOLING SYSTEM CONTROL

Thermostat T_1 measures the outdoor temperature and thereby automatically determines the inside dry-bulb temperature control point. Thermostat T_2 in the conditioned space measures the temperature of that space and controls the refrigerant to the cooling coil so as to maintain the temperature in the space being conditioned at the point which has been set up by thermostat T_1.

It is usually found desirable to adjust the indoor temperature between available limits with the outdoor temperature all of which is fully described in Chapter 3. Various combinations of control may be applied to cooling systems to secure desired relationship between outdoor temperature and resultant indoor temperature and humidity.

All Year Systems

An all year central fan conditioning system consists of the combination of a ventilating system and a cooling system.

During certain seasons of the year, it is sometimes possible to control

the dew-point of the air discharged from an air washer by regulating the relative quantities of outdoor and return air. The use of this method for controlling the outdoor and return air dampers may also provide for automatic change-over from the heating to cooling cycles, providing thereby for the maintenance of a fixed dew-point temperature in the air-discharge during both cycles.

Complete automatic control of all year systems incorporates an automatic change-over between the cooling and heating cycles. If the installation necessitates operation of manual switch or other device to change over between the heating and cooling cycles, then the control system is semi-automatic. The full automatic change-over between cycles becomes particularly desirable in the early and late portions of the cooling and heating seasons when heating is required during the early and late portion of the day and cooling may be required during the middle of the day.

A system for the control of an all year conditioning system providing for automatic change-over from the cooling to heating cycles is illustrated in Fig. 6.

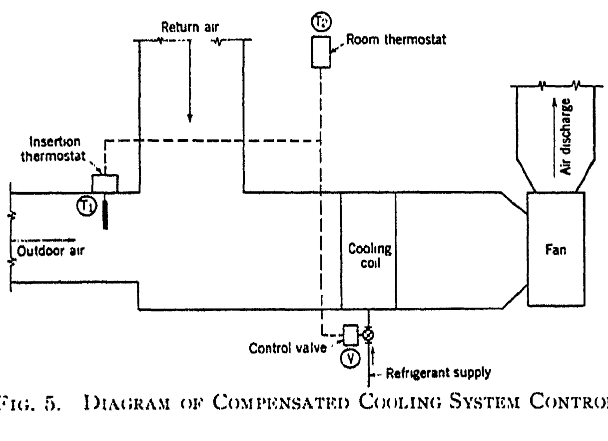


FIG. 5. DIAGRAM OF COMPENSATED COOLING SYSTEM CONTROL

During the heating cycle, thermostat T_1 in the return air or room measures the temperature of the conditioned space and modulates control valve V_1 which, in turn, modulates the flow of steam to the heating coil so as to maintain a fixed temperature in the space. Humidity control H_1 measures the relative humidity in the space being conditioned and opens control valve V_2 so as to admit water to the sprays whenever moisture is required in the air.

During the cooling cycle, thermostat T_2 in the return air measures the temperature in the space being conditioned and modulates control valve V_3 which, in turn, modulates the flow of water to the cooling coil so as to maintain a fixed temperature in the space. Humidity control H_2 measures the relative humidity in the space being conditioned and then assumes command of control valve V_3 whenever the relative humidity exceeds a predetermined amount.

During the heating cycle thermostat T_3 acts as a low limit. It assumes command of control valve V_1 whenever it is necessary to prevent the air

discharge temperature from falling below a minimum point. Thermostat T_3 may also be arranged to act as a low limit during the cooling cycle if the conditions of the installation make it desirable.

Thermostat T_4 installed in the inlet to the conditioner controls damper motor M_1 which in turn regulates the relative quantity of outdoor and return air admitted to the system. This damper action may be provided with a minimum setting of the outdoor air damper so that a minimum fixed requirement of outdoor air will be insured for ventilating purposes.

Humidity control H_3 measures the outdoor air relative humidity and prevents the outdoor air damper from opening beyond its minimum

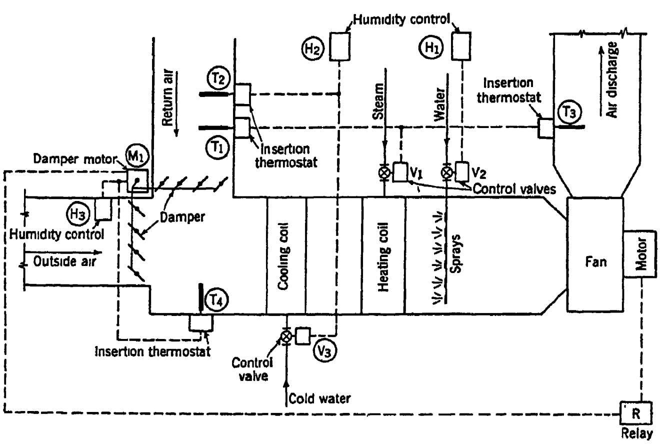


FIG. 6. DIAGRAM OF COMPLETE AUTOMATIC CONTROL ALL YEAR AIR CONDITIONING SYSTEM

position whenever the outdoor air relative humidity exceeds a predetermined point.

When the fan is stopped, relay R positions damper motor M_1 so as to close the outdoor air damper.

Thermostat T_1 must be set at a lower temperature than thermostat T_2 in order that each may assume command upon the fall or rise respectively of the temperature of the return air. As an example, T_1 might be set at 72 F and T_2 at 76 F. When the temperature of the return air approaches 72 F, it would indicate that a change had taken place from the cooling to the heating cycle and when the return air approaches 76 F, it would indicate that a change has taken place from the heating to the cooling cycle.

UNIT SYSTEMS

A unit system provides for the same functions as a central fan system except that the actual conditioning is usually done within the space being conditioned instead of at some central location outside of the space. The

automatic control problems, therefore, become exactly the same as for central fan conditioning systems except that compactness, ease of installation and control cost often assume somewhat more importance.

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat the function of which is to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan at the command of the thermostat would cause objectionable drafts. To prevent this occurrence, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by the temperature of the steam or return pipe or the pressure of the steam supply.

In some cases it is desirable to operate the unit heaters continuously for circulation of air where, due to the type of installation, cold drafts will not result therefrom. In such instances the room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

Cooling Units

The recommended form of temperature control for the cooling unit contemplates the continuous operation of the cooling unit fan with automatic two position regulation of the compressor or cooling coil as determined by a room thermostat or by a temperature controller measuring the temperature of the return air as it is taken into the cooling unit. Such operation insures continuous circulation of the air in the room served by the cooling unit, and in addition to providing the cooling effect due to the moving air, this circulation overcomes the tendency of air to stratify. Thus, as this temperature tends to rise, the temperature controller will open the valve supplying either refrigerant or cold water to the cooling unit coil or start the compressor.

Cooling units may also be controlled by arranging the room thermostat to start and stop the fan motor or by a combination of motor and refrigerant control.

A humidity controller may be used in conjunction with the thermostat as a high limit control to permit the cooling and dehumidifying of the air

whenever the relative humidity rises above some predetermined point such as 60 per cent even though the thermostat is satisfied. This control is desirable on damp days or in conditions where the humidity load may become excessive, but its operation will result in excessive cooling unless some means of reheating is provided.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

In the application of control to unit ventilators the essential requirement is that the action be graduated to prevent sudden changes in the temperature of the discharged air and where direct radiation is used in conjunction with the unit that the cycle of control be so arranged that steam will be admitted to the direct radiation only when the unit is unable to carry the heating load. This arrangement prevents the unit from delivering air at low temperatures to offset the overheating effect of the direct radiation and results in the delivery of a higher percentage of tempered air.

There are two general types of control applied to unit ventilators as follows:

1. The mixing or by-pass damper type of unit is provided with a damper, equipped with a damper motor, which, under control of the thermostat, passes air through and around the heating element in such proportion as to maintain a uniform room temperature, the two streams of cold and tempered air being mixed and diffused at the ceiling. A control valve may also be used on the steam supply to the heating element of the unit and should be arranged to throttle the steam supply when the damper approaches a position to by-pass all of the air.

The outside air damper of this type of unit is usually provided with a damper motor and controlled by a remote manual switch to assume either a fully open or fully closed position.

2. The recirculating type of unit ventilator is equipped with a control valve on the steam supply to the heating element of the unit and with a damper motor on the outside air-recirculating air damper, both under the control of the room thermostat. Some units are so arranged that a mixture of outside air and recirculated air passes through the heating element and others so that only the recirculated air is heated.

The fundamental requirements of control as applied to this type of unit is that the steam supply to the direct radiation, the steam supply to the unit ventilator and the mixing of outside and recirculated air be accomplished in a definite cycle or sequence to meet the requirements of the particular unit used and differs from the mixing damper type of unit in that the percentage of outside air and recirculated air delivered by the unit is determined by room temperature. The damper motor is sometimes arranged so that a fixed minimum quantity of outside air is delivered continuously as soon as the room has reached a predetermined temperature. A limit thermostat, either in the mixing chamber or in the discharge of the unit, is sometimes used in conjunction with the room thermostat, so arranged that the action on either the control valve or the dampers, or both, is stopped when a predetermined minimum temperature has been reached in the unit discharge, to prevent delivery of air at a lower temperature.

For additional information on the control of unit ventilators when installed and operated under various types of applications refer to Chapter 23.

All Year Conditioning Units

It is desirable to provide for automatic change-over between the cooling and heating cycles in the control system for all year conditioning units because of the probable necessity of changing over a large number of units if done manually.

A control system for an all year conditioning unit providing for the automatic change-over is shown in Fig. 7. Operation of the control equipment is as follows:

1. *During the Heating Cycle.* Combination controller T_1 measures the temperature in the space being conditioned and opens control valve V_2 so as to admit steam to the heating coil whenever heat is required so as to maintain a fixed temperature in the space. Combination controller T_1 also measures the relative humidity in the conditioned space and opens

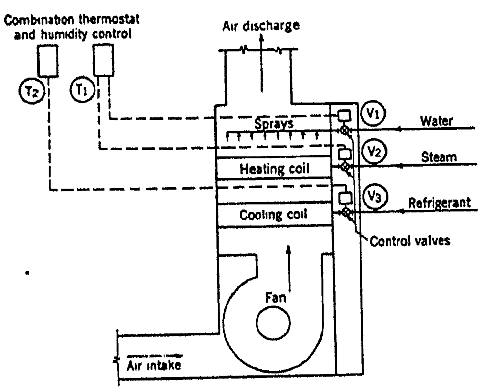


FIG. 7. ALL YEAR AIR CONDITIONING UNIT WITH COMPLETE AUTOMATIC CONTROL

control valve V_1 so as to admit water to the sprays whenever moisture is required in the space.

2. *During the Cooling Cycle.* Combination controller T_2 measures the temperature and humidity in the conditioned space and opens refrigerant control valve V_3, thereby admitting refrigerant to the cooling coil whenever cooling is required to maintain the temperature or relative humidity within predetermined maximum limits.

The temperature control point of controller T_1 must be set at a lower point than that of controller T_2 in order to provide for the automatic change-over between the cooling and heating cycles. As an example, controller T_1 might be set at 72 F and controller T_2 at 76 F. As the temperature in the space approaches 72 F, it would indicate a change from the cooling to the heating cycle and when the temperature in the space approaches 76 F, it would indicate a change from the heating to the cooling cycle, and the corresponding controllers would assume command. In the same way, the relative humidity control point of controller T_1 would be set at a lower point than that of controller T_2. As an example, T_1 might be set at 35 per cent and T_2 at 60 per cent.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant. There are many types of burners and many types of automatic control, and it is essential that the proper type of control equipment be selected to fulfill the requirements of the burner equipment and its application.

Combustion regulation equipment should be used on the larger commercial and industrial applications to control the secondary air supply and thereby provide for economical operation. This type of control will usually consist of a pressure regulator which measures and controls the pressure over the fire and which thereby indirectly regulates the carbon dioxide percentage in the flue gas.

On all automatically-fired steam boilers it is advisable to provide control equipment which will stop the burner operation in case the boiler water line falls below a predetermined level of safety.

Thermostats used to control automatic fuel appliances may be provided with clock mechanisms which will operate to maintain lower temperatures during night hours for economy of fuel.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. For purposes of preventing abnormally high temperatures in the bonnet of gas-fired furnaces or in the temperature of

the water in gas-fired hot water heating boilers or excessive pressures in gas-fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory and economical operation, all automatically fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

At certain seasons of the year, the operation of the stoker under the requirements of the thermostat may be so infrequent that there is a possibility of the fuel in the combustion chamber burning out or the fire going out between operations. To prevent this occurrence, automatic controls may be utilized to operate the stoker independently of thermostat requirements, sufficiently to sustain the fire either through a timing device functioning for short periods at predetermined intervals or through a temperature control device reacting to minimum stack or boiler temperatures. Control may also be utilized to prevent stoker operation and the delivery of coal into the combustion chamber in the event that the fire has gone completely out. This control is governed normally by the stack temperature and shuts down the stoker after a predetermined minimum stack temperature is reached.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are normally equipped with single room thermostat, limit and safety controls as outlined above under Control of Automatic Fuel Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air

furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when temperature or pressure conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system, or controls the operation of a circulating pump in a forced circulation hot water system.

Air Conditioning Systems

Residential air conditioning systems are of various types normally including a heating source and a motor-driven fan for circulating air. In addition, such installations may involve spray-head equipment, the purpose of which may be only to supply humidity, or which, in some instances, are of greater capacity and serve not only to humidify but to wash the air passing through them. It is also common practice to include dry filters to aid in air cleaning. Such installations distribute suitably heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if the washer unit is supplied with water at suitable temperature or if such an installation is equipped with other refrigeration means.

During the heating cycle the regulation of temperatures is normally one or the other of the problems previously discussed in connection with the various types of heating sources described, such as the oil burner, gas burner, stoker or the coal-fired heating plant under automatic control. Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

When cooling equipment is provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

The selection of automatic control equipment for residential air conditioning systems is just as important as for commercial installations. Fewer controls are generally used and systems are usually less complicated except in the case of a very large residence installation when the control system may become as complete as the commercial installation.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Compressor Type Refrigeration

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with either a temperature or pressure controller provides a safety feature against the development of excessive pressures on the high side of the compressor.

Refrigeration by Ice

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

Vacuum Refrigeration

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two position or positive control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Refrigeration by Well Water

When well water is available in sufficient quantities at low temperatures during the cooling season, it may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

INDUSTRIAL PROCESSES

There are many industrial processes requiring automatic temperature and humidity regulation. The control equipment operates on the same principles that have been described, but it is often especially designed for each particular process. Each installation, or the installation for each process, is likely to be a problem peculiar to that process.

PROBLEMS IN PRACTICE

1 ● What important functions of heating, ventilating, and air conditioning systems do automatic controls fulfill?

Controls are applied to maintain adequate requirements for human comfort and efficiency; to maintain requirements for industrial processes; to obtain economy in operation; and to provide necessary safety measures.

2 ● How may temperature control be obtained in a room heated by a unit heater?

With constant steam supply, the unit heater motor may be started or stopped by a thermostat, either directly or through a relay. With intermittent steam supply, operation of the motor by thermostat can be limited to the time that steam is available, by using a reverse-acting temperature or pressure limit switch.

3 ● How may temperature control be obtained in a room cooled by a self-contained mechanical unit?

The fan operation may be controlled by a manual switch, while a room thermostat in conjunction with a solenoid valve may regulate the flow of the refrigerant to the coil. The thermostatic circuit might be operative only when the fans are running; and the compressor might be controlled by refrigerant pressure.

4 ● How may temperature control be obtained in a room heated by an automatically-fired warm air furnace?

A room thermostat might control the combustion unit; and a limit switch in the top of the furnace unit, when at a low setting of its control might operate the fan whenever there is a rise of temperature, and when at a high setting of its control it might shut off the combustion unit. A room humidity control operating a solenoid valve on the water supply to the humidifier, or operating a relay on the recirculating pump motor to the humidifier, may be connected in parallel with the fan motor. Humidification may be supplied only when heat is supplied and when the humidity control acts in conjunction with a time switch.

5 ● How may humidity be controlled in a unit humidifier for a steam or hot water heating plant?

Since heat is required for evaporation, a temperature limit switch, preferably of the immersion type, may be placed in the heating supply riser to cause the unit to be inoperative when heat is not available. A room humidity control will operate a solenoid valve on the water supply to the sprays. Both the solenoid valve and the humidity control may be electrically wired in parallel with a fan motor, and be subject to the temperature limit switch.

Chapter 38

MOTORS AND CONTROLS

Direct Current Motors, Alternating Current Motors for Single Phase and Polyphase, Special Applications, Classification of Motors, Manual Control, Automatic Control, Pilot Controls, Direct Current Motor Control, Squirrel Cage Motor Control, Multispeed Motor Control, Slip Ring Motor Control, Single Phase Motor Control

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors used for heating, ventilating and air conditioning applications may be divided into two general classifications as follows:

1. For use with direct current.
2. For use with alternating current.

DIRECT CURRENT MOTORS

There are three types of direct current motors available:

1. Shunt Wound.
2. Compound Wound.
3. Series Wound.

Shunt Wound motors being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of applications in the field of heating, ventilating and air conditioning. They may be used on reciprocating pumps and compressors, if started under unloaded conditions.

Compound Wound motors are required for application to compressors, stokers, reciprocating pumps when started under loaded conditions, and also when applied to similar equipment where high starting torque is required. Whenever frequent starting makes high starting and accelerating torque desirable, or where sudden changes of load are encountered, compound wound motors are used.

Series Wound motors find only limited application in a few special cases and are available in only a limited range of sizes.

Speed Characteristics

Direct current motors are available with speed characteristics of four types:

1. Constant speed.
2. Adjustable speed.
3. Adjustable varying speed.
4. Varying speed.

Constant Speed motors may be shunt wound or compound wound. Shunt wound motors have a nearly flat speed-load characteristic, with a regulation of 15 per cent for up to ¾ hp, 12 per cent for one to 5 hp and 10 per cent for 7½ hp and larger, based on full load speed.

Compound wound motors have a speed regulation over the range from full load to no load of not more than 25 per cent, based on full load speed.

Adjustable Speed motors are usually shunt wound since it is impractical to maintain the proper relation between the shunt and series fields of compound wound motors when wide variations of the field strength are required to obtain the speed adjustment.

Adjustment of the speed of shunt wound motors is obtained by field control on motors rated at ¾ hp and larger, with the minimum or base speed at full field strength and higher speeds at reduced field strength (obtained by adding resistance in the field circuit). The speed regulation from no load to full load will not exceed 22 per cent for 2 to 5 hp; nor 15 per cent for 7½ hp and larger. Below 2 hp, the regulation may exceed 22 per cent. If closer speed regulation is required, specifically wound motors must be obtained.

Practically constant horsepower output is obtained at all speeds up to a ratio of 2 to 1. For higher speed ratios, the horsepower rating at the minimum speed is less than at the maximum speed, this difference varying with the speed ratio. High efficiency is maintained over the entire speed range. Most listed constant speed motors are suitable for operation up to a speed ratio of 2 to 1 by the use of proper control equipment.

Adjustable Varying Speed motors may be either shunt or compound wound and speed adjustment is obtained by adding resistance in series with the armature. The speed thus obtained is always below the rated full-field speed. Any standard shunt or compound wound constant speed motor may be used in conjunction with the proper armature resistor. The usual range of speed reduction is 50 per cent. The speed obtained for any setting of the resistor depends on the load of the motor and will vary with this load.

The speed regulation at high speed is comparable to a constant speed motor, but becomes poorer as the speed is decreased.

When operating at reduced speed, an increased torque requirement which the motor could easily handle at rated speed is easily sufficient to stall the motor; for example, a motor operating at two-thirds speed would be stalled by a torque about 50 per cent in excess of the normal requirement.

The efficiency of the motor is reduced as the speed is reduced, since the

loss in the resistor is greater at lower speeds. Speed reduction by armature control is usually selected where:

1. A wide speed range is not required.
2. Close speed regulation is not necessary.
3. Operating time at reduced speed is short.
4. Operating load at reduced speed is small so that the reduced efficiency can be ignored.
5. The rating is less than 1 hp.

Varying Speed motors are series wound and the speed varies with the load on the motor. They should be used where:

1. The load is practically constant or increases with speed.
2. The motor can easily be controlled by hand.

They should not be used where there is a possibility of operation without load or at a reduced load, as the speed of the motor may become dangerously high.

For shunt wound motors with full field strength, the starting torque varies almost directly with the starting current, which is dependent on the resistance in the armature circuit. With varying positions of the starting rheostat, it is possible to obtain a wide range of starting torque, within the limits of starting current permitted by the power company.

A compound wound motor requires somewhat less current for the same starting torque. The maximum torque of shunt, series, and compound wound motors is limited by commutation.

ALTERNATING CURRENT MOTORS

Alternating current motors may be divided into two main groups, namely, (1) those operating on single phase current, and (2) those operating on polyphase current.

1. Single phase motors are available in four common types:
   a. Capacitor motors.
      1. Full capacitor.
      2. Capacitor start-induction run.
   b. Repulsion induction motors.
   c. Repulsion start, induction run motors.
   d. Split phase motors.
2. Polyphase (2 or 3 phase) motors are available in four common types:
   a. Squirrel cage induction motor.
   b. Automatic start induction motor.
   c. Slip ring, wound rotor induction motor.
   d. Synchronous motor.

Where the public utility supplying the current determines that a particular installation should be served with polyphase current, it is generally understood that the major portion of the motors will be for polyphase current, although it is commonly acceptable for the smaller motors to be single phase. This will limit the use of single phase current to the smaller motor ratings and the polyphase to the larger motors. Domestic and semi-commercial installations will invariably be single phase.

Single Phase Motors

Capacitor type motors are available in ratings up to 10 or 15 hp for general purposes. These motors are recommended for pumps, compressors and fan duty including housed centrifugal fans and propeller fans. The general purpose motor is commonly known as a high torque capacitor motor having approximately 300 per cent starting torque with normal current and having a different value of capacitance for starting and running which is automatically changed over by a mechanical or electrical means.

Capacitor motors for *fan duty* are usually divided into the open high torque type for belted fans and the totally inclosed non-ventilated low torque type for propeller fans mounted directly on the motor shaft. The open low torque capacitor motor may be used with small centrifugal fans mounted on the motor shaft.

Although the motors for *belted fans* are called high torque, the available starting torque is somewhat less than the torque of the general purpose motor and the slip at full load is approximately 8 per cent. With this larger amount of slip, adjustable speed down to 60 or 70 per cent of rated speed may be obtained by line voltage variation. Motors for *propeller fan* drive may be supplied with sleeve bearings to obtain greater quietness in the smaller sizes where the fan thrust does not exceed approximately 25 lb. For larger fans, thrust ball bearing motors should be used. Low torque capacitor motors have approximately 50 per cent starting torque and do not change the value of capacitance from start to run.

Capacitor motors with *high slip* may have taps brought out from the main winding which when connected to the line, give a second speed of from 65 to 70 per cent of the normal speed. This type of motor must be specially designed for the individual fan, otherwise the correct low speed will not be obtained. Care should be exercised in applying it to centrifugal fans where restriction to the air flow through the use of adjustable dampers changes the motor load and consequently the speed. This same effect is also found in transformer speed controllers, however, a series of transformer taps allow for a selection which partially overcomes the effect of change in motor load.

Capacitor start-induction run motors are usually confined to the smaller horsepower ratings and differ from the capacitor motors by having no running capacitor. The value of starting capacitance used may vary with the different types of applications involved. These motors may be used for practically any of the applications met in air conditioning. However, consideration should be given to the fact that they are not as quiet as a capacitor motor.

Repulsion induction motors start as repulsion motors and operate under full speed as combined repulsion and induction motors through the inherent characteristics of the motor which has, in addition to the wire winding with commutator, a buried squirrel cage winding. No additional switching devices are required to change over from start to run. This and the repulsion motor described below may be used for constant speed drives where high starting torque is required and where commutator and brush noise is not a factor.

The *repulsion start-induction* run motor starts as a repulsion motor, has a switching means for transferring from start to run which short circuits the commutator and permits operation under full speed as a wound induction motor. This motor is suitable for applications similar to those for which the repulsion induction motor is used.

The *split phase* motor has a high resistance auxiliary winding in the circuit during starting which is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions, it operates as a single phase induction motor with one winding in the circuit. These units are available for the lower horsepower ratings and when equipped with a high slip rotor may be used for adjustable varying speed through line voltage control.

Polyphase Motors

Squirrel cage induction motors are available in three types and a full range of sizes:

1. The normal torque, normal starting current squirrel cage motor has close speed regulation, high efficiency, high power factor, medium starting torque, high pull-out torque, and is suitable for general purpose applications. This motor has a large current inrush and a low starting current power factor. It operates with these characteristics only when started directly across the line on full voltage. When central stations require current limiting starting equipment on such motors, the starting torque is less. Current limiting hand operated starters are standard equipment.

2. The normal torque, low starting current squirrel cage motor has approximately the same torque as the normal current motor, but the starting current is about 20 per cent less than the normal torque motor on full voltage and ordinarily within the *National Electric Light Association* locked rotor current limits on sizes up to 30 hp.

   This motor lends itself to automatic or remote control because no current limiting starting equipment is necessary up to and including 30 hp. A magnetic starter with low voltage and thermal relay overload protection gives the most satisfactory service.

3. The high torque, low current squirrel cage motor has a starting torque approximately 25 to 50 per cent greater than the normal torque motor on full voltage with starting current approximately 10 per cent less than the normal torque motor started on full voltage, but within the required limits on 30 hp sizes and smaller. These motors are also started directly across the line on full voltage through a magnetic starter or other approved starting device.

These three types of motors are also available in two, three, or four speed designs with variable torque, or constant torque characteristics. Two speed motors may be either single, or two winding; three speed motors are single, two, or three winding; and four speed motors are two, three, or four winding. When a motor is wound with a winding for each speed, better operating characteristics may be obtained because no sacrifice is made for the other speed and operating characteristics approaching single winding motors may be expected.

Frequently, multispeed motors lend flexibility to an installation that cannot be obtained in any other way.

Multispeed motors are started directly across the line through magnetic starting equipment with overload and low voltage protection and compelling relays to insure starting on low speed regardless of the ultimate running speed. Starting on low speed limits the starting current to the starting current of the low speed winding and consequently lowers the maximum demand.

TABLE 1. CLASSIFICATION OF MOTORS

Current	Type	Speed Characteristics	Full Voltage: Starting Torque	Full Voltage: Starting Current	Hp Range	Type of Application See Footnote*
		Constant Speed Drives				
Direct	1. Shunt	Constant	Medium	Medium	All	(*a*) Fans and (*c*) Centrifugal Pumps
Direct	2. Compound	Constant or Variable	High	Medium	All	(*b*) (*c*) (*e*) Reciprocating Pumps and frequent or hand starting
Direct	3. Series	Variable	High	Medium	Small	(*d*) Fans direct connected
Poly-phase	4. Squirrel Cage General Purpose	Constant	Normal	High 6-8 Times	All	(*a*) Fans and (*c*) Centrifugal Pumps
Poly-phase	5. Squirrel Cage Medium Torque	Constant	Normal	Medium 5-6 Times	Medium Small	(*a*) Fans and Centrifugal Pumps
Poly-phase	6. Squirrel Cage High Torque	Constant	High	Medium 5-6 Times	Medium Small	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
Poly-phase	7. Automatic Start High Torque	Constant	High	Low 3 Times	Medium	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
Poly-phase	8. Slip Ring Wound Rotor	Constant	High	Low 1-3 Times with secondary control	All	(*a*) and Hoists (*b*) Reciprocating Pumps (*c*) and Frequent (*e*) or Hand Start
Poly-phase	9. Synchronous High Speed	Constant	Medium	Medium 5-7 Times	Medium Large	(*a*) Fans and Centrifugal Pumps
Poly-phase	10. Synchronous Low Speed	Constant	Low	Low 3-4 Times	Medium Large	(*a*) Reciprocating Compressors Starting Unloaded
Single phase	11. Capacitor	Constant	High	Normal	Medium Small	(*b*) Pumps and Compressors

*Applications:

a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded.

b. Drives having high starting torques, such as reciprocating pumps and compressors started loaded.

c. Similar to (*a*) except where frequent or hand starting (large WR^2) requires a higher starting and accelerating torque.

d. Fans direct connected.

e. Stoker drives.

TABLE 1. CLASSIFICATION OF MOTORS—(Continued)

CURRENT	TYPE	SPEED CHARAC-TERISTICS	FULL VOLTAGE STARTING TORQUE	FULL VOLTAGE STARTING CURRENT	HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
SINGLE PHASE	12. Capacitor Fan	Constant	High	Medium	Medium Small	(*a*) Fans—belted
	13. Capacitor Fan	Constant	Low	Medium	Medium Small	(*d*) Fans—direct
	14. Capacitor Start Induction Run	Constant	Any	Medium	Medium Small	(*a*) Fans (*b*) Pumps and Compressors
	15. Repulsion Induction	Constant	High	Medium	Medium Small	(*a*) Fans (*b*) Pumps and Compressors
	16. Repulsion Start Induction Run	Constant	High	Medium	Medium Small	(*a*) Fans (*b*) Pumps and Compressors
	17. Split Phase	Constant and Adjust-table	Medium	Medium	Frac-tional	(*a*) Fans (*b*) Pumps and Compressors
		Adjustable Speed Drives				
DIRECT	18. Shunt Field Adjustment	Constant	Medium	Medium	All	(*a*) Fans and (*e*) Centrifugal Pumps
	19. Shunt Armature Resistor	Variable	Medium	Medium	All	(*a*) Fans and (*e*) Centrifugal Pumps
POLY-PHASE	20. Squirrel Cage High Slip, Tapped Winding	Variable	Medium	Medium	Medium Small	(*a*) Fans
	21. Squirrel Cage High Slip, Trans-former Adjust-ment	Variable	Medium	Medium	Medium Small	(*a*) Fans
	22. Squirrel Cage Separate Wind-ing or Regrouped Poles	Constant Multi-Speed	Medium or High	Low	All	(*a*) Fans (*b*) Pumps and (*c*) Compressors

TABLE 1. CLASSIFICATION OF MOTORS—(Continued)

CURRENT	TYPE	SPEED CHARACTERISTICS	FULL VOLTAGE STARTING TORQUE	FULL VOLTAGE STARTING CURRENT	HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
POLY-PHASE	23. Wound Rotor, Slip, Ring, External Secondary Resistance	Variable	High	Low	All	(a) Fans and (b) Centrifugal Pumps
SINGLE PHASE	24. Capacitor High Torque Tapped Winding	Variable	High	Normal	Medium Low	(a) Fans, belt
	25. Capacitor Low Torque Tapped Winding	Variable	Low	Medium	Medium Low	(d) Fans, direct
	26. Capacitor High Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	27. Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	28. Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(d) Fans

Often where the central station requires current limiting starting equipment for the normal torque, normal starting current motor, it is advisable to use the normal torque low starting current multispeed motor.

High slip polyphase motors may be used for adjustable varying speed drives in a manner similar to that described for capacitor motors, with either a transformer speed regulator or tapped motor windings.

It is apparent from these motor characteristics that a squirrel cage motor may be selected for operating any air conditioning and allied equipment.

Automatic start induction motors are constructed with two windings on the rotor, one of which is a high resistance, squirrel cage winding used in starting and gives a high starting torque approximately the same as the high torque, squirrel cage. A centrifugal mechanism within the motor switches to the second low resistance winding when the motor comes up to speed, thus obtaining running characteristics equal to the normal torque, normal current squirrel cage motor. The power factor of the starting current is high.

Slip ring wound rotor motors are built for two classes of service, constant speed and adjustable variable speed. The motors are identical in each case and use the same primary control, the only difference being in the secondary control.

Slip ring motors for constant speed service are used where high starting torque with low starting current is required for bringing heavy loads up to speed. The resistance is in the secondary or rotor circuit, only when starting, and is short circuited when the motor is up to speed.

For adjustable varying speed service, part or all of the secondary controller resistance is in the circuit whenever the motor is operating below full speed. The speed obtained with a given resistance in the secondary circuit is dependent on, and changes with the load on the motor. The horsepower developed by the motor is approximately proportional to the speed, whereas the power required by the motor is practically the same at reduced speed as at full speed, hence the efficiency at reduced speeds is much lower than at full speed.

Synchronous motors are ordinarily used only where there is a need for, or advantage in, obtaining power factor correction. It is necessary to consider each application as a special case which must be individually engineered, since for satisfactory operation, the combined moment of inertia of the compressor fly wheel and motor rotor must be correctly established.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 1.

SPECIAL APPLICATIONS

A few applications of motors may require special constructions such as splash proof, explosion proof, fully enclosed, and self-ventilated to meet hazardous or special duty conditions. These requirements are frequently encountered in certain industrial applications, in which cases it is necessary to select the motors from the viewpoint of service conditions, as well as the required operating characteristics to meet the demands of the machines being driven.

CONTROL EQUIPMENT FOR MOTORS

In selecting control for alternating and direct current motors it is necessary to determine whether the installation is to be operated by manual or automatic control. The available controls and the function of each group of apparatus may be outlined as follows:

1. Manual Control:
   a. To establish current.
      (1) Snap switch.
      (2) Knife switch.
      (3) Manually operated contactor.
      (4) Drum switch.
   b. Establish current and add overload protective device.
      (1) Snap switch with overload element.
      (2) Knife switch with fuse or thermal cutout.
      (3) Manual contactor with overload protective device; also reduced voltage starting compensator.
      (4) Drum switch with overload protection.
   c. Establish current and add overload and low voltage protective devices.
      (1) Not used.
      (2) Not used.

(3) Manual contactor or reduced voltage compensator with overload and low voltage release.
(4) Drum switch equipped with latch coil to give low voltage release.

2. Automatic Control:
   a. To start on full voltage.
      (1) Without overload device.
      (2) With overload device.
      (3) With combination overload device and knife switch.
   b. Reduced voltage starting.
      (1) Primary resistance type starter.
      (2) Auto compensator type.
      (3) Reactance type.

PILOT CONTROLS

In selecting pilot control devices to operate in conjunction with either manual or automatic motor control, it is necessary that they be classified as follows:

1. *Two Wire Control.* Most thermostats, float switches, and pressure regulators, provide two wire control which gives low voltage release. A three position pilot switch can be used in connection with this method and thus provide manual control. With a low voltage (12 or 20 volt) control circuit it is desirable to use a low voltage thermostat. When this type of thermostat is used it will be found that a saving in the wiring cost results. When using the low voltage thermostat on a control circuit a relay and transformer panel should be used instead of the low voltage coil on the starter.

2. *Three Wire Control.* Momentary contact start and stop push button stations are usually furnished as standard accessories with automatic starters, which gives low voltage protection. This control cannot be used in combination with two wire pilot devices.

In selecting manual control for an alternating or a direct current motor, the common practice is to locate the control near the motor. When the control is installed at the motor, an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Frequently manual control is employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system In other cases manual control in the form of drums, when used with multispeed motors, is only used as a speed setting device with the starting and stopping functions operated automatically through thermostats, and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is being operated on automatic control with less dependence on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere. In the majority of air conditioning installations, requiring motors 1 hp and larger, two or three phase alternating current is usually supplied.

DIRECT CURRENT MOTOR CONTROLS

Air conditioning installations using direct current power are now only used where alternating current is not available. Direct current motors are always started through starters, which are devices using a resistance to be put in series with the armature circuit during starting only, the resistance being gradually cut out as the motor comes up to speed. The starting current is held within safe limits by the use of the resistance.

The speed of a direct current motor may be regulated by the following methods:

1. Speed regulation by field control—by using a device with resistance to be put in series with the field winding. After the motor has been started to be used to increase the speed of the motor above full field speed.
2. Speed regulation by armature control—by using devices with resistance to be put in series with the armature circuit to be used to reduce the speed of the motor below full field or normal speed.
3. Combinations of field and armature control, so that the starting, field control, or armature control may be combined in a single unit.

Field control is usually preferred, depending on the size of the installation. For example, if a direct current motor were required with speed regulation between 1200 and 600 rpm, a choice of supplying a 1200 rpm motor with armature control or a 600 rpm motor with field control, both giving the same speed variation would be possible. While the 1200 rpm motor with armature control is lower in first cost than the 600 rpm motor with field control, the cost of operating the 600 rpm motor with field control is less and will save the difference in first cost over a period of time depending on the size of installation. A wide speed variation can be easily obtained in a direct current motor by using a combination of field and armature control.

SQUIRREL CAGE MOTOR CONTROL

To meet the requirements of various drives of an air conditioning system, three types of squirrel cage, two or three phase motors may be used:

1. Normal torque, normal starting current.
2. Normal torque, low starting current.
3. High torque, low starting current.

Because of the large current inrush of the normal torque, normal starting current motor, central stations usually require current limiting starting equipment on such motors above 5 hp. To meet the starting current requirements, manual or automatic current limiting starting compensators are used. These compensators are equipped with 50, 65 and 80 per cent voltage taps, the 65 per cent tap being regularly furnished when the compensator leaves the factory. Motors 5 hp and smaller have starting currents within the requirements of central stations and manual or magnetic, full voltage control may be used.

The normal torque, low starting current motor has a starting current which is approximately 20 per cent less than the normal current motor on

full voltage and well within the required current limits on 30 hp sizes and smaller. This motor, therefore, lends itself to across-the-line control because no current limiting equipment is necessary. In selecting motors for fans, pumps, or blowers, it should be noted that while the cost of the normal starting torque, low starting current motor is higher, the cost of full voltage control is lower, so that the total cost of low starting current motors with across-the-line control is lower.

A magnetic starter with low voltage and thermal overload protection gives the most satisfactory service. These switches may be controlled by remote push button stations, thermostats, or pressure switches to meet the requirements of any particular installation.

The high torque, low starting current motor has a starting current approximately 10 per cent less than the normal torque, low starting current motor when started on full voltage. These motors, most commonly used on compressor drive, can be started directly across-the-line with manual or magnetic starters.

Adjustable varying speed motor control by terminal voltage regulation requires a tap-changing switch manually or magnetically operated. Such a control switch operates to alter the voltage applied to the motor by contacting different auto-transformer voltage-ratio taps or by changing the amount of resistance inserted in the primary or line circuit.

MULTISPEED MOTOR CONTROL

To make an installation more flexible, multispeed motors are available with two, three or four speed designs, with variable torque, constant torque or constant horsepower characteristics. Multispeed may be started by means of manual or magnetic starting equipment.

When using automatic magnetic control with two, three, and four speed separate winding or consequent pole motors, control is obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button, which is known as selective speed control. It is commonly used in the smaller theatre installations where the fan and motor is located backstage and the speed control is located in the lobby.

Magnetic multispeed motor controllers may also be provided with a compelling relay which makes it necessary that the operator press the first speed button before regulating the motor to the desired speed. This assures the operator that the motor is always started at low speed before the motor is adjusted to one of the higher speeds. Starting on low speed limits the starting current to the starting current of the low speed winding, and therefore, permits the use of motors in sizes larger than ordinarily permitted by central stations for full voltage starting.

Timing relays, which provide for automatic acceleration, may be used for control. With the automatic acceleration feature, it is only necessary to press the button for the desired speed. The motor will always start in low speed and automatically step up to the desired speed.

Where the change of speeds does not occur at regular intervals, and where it is only necessary to change from one speed to another to take

care of seasonal requirements, a manual drum speed selector may be used. This drum is used to select the proper motor speed while an automatic starter is used to start and stop the motor.

The smaller size speed selector drums rated 10 hp at 220 volts and smaller may also be used as a motor starter to make and break the current, as well as, serving as a speed selector device. Reversible or non-reversible drums may be supplied depending on the requirements of the installation.

In the large size drums, a separate contactor must be provided to make and break the current. The contactor may be any approved starter. Overload and low voltage protection may be accomplished by using a magnetic starter. No push button station is required, the handle switch on the drum having the same characteristics as a three wire push button station.

In selecting two speed motors for fan, pump, blower, or compressor drive it will be found that the two winding motors are more expensive than the single winding. The control for two speed, two winding motors is more economical and the combined price of the motor and contactor is only slightly higher. Because of the better performance of the two speed motor and the factor of safety in having two independent motor windings, the increased cost is considered worth the difference.

SLIP RING MOTOR CONTROL

When close speed regulation and low starting current is required slip ring or wound rotor motors are used. Slip ring motors are built for two classes of service, constant speed and adjustable varying speed. The motors for the two classes of service are identical, the only difference being in the secondary control used with the motors. Control for both primary and secondary of a slip ring motor is required.

The primary control for a constant or adjustable speed is the same type as used with squirrel cage motors. Manual or magnetic starters, across-the-line type, may be used depending on the installation.

The starting current and starting torque of a slip ring motor are almost entirely dependent on the amount of resistance in the secondary control and in the manner in which the secondary control is operated. The *National Electric Manufacturers Association* has adopted service classifications which allow a selection of resistors permitting a starting current on the first contact of resistance varying from approximately 25 per cent of full load current to approximately 200 per cent of full load current or more, and permitting the resistor to remain in the secondary circuit of the motor for a period varying from not more than 15 seconds during an interval of operation from 4 minutes to continuous.

Speed regulation of a slip ring motor is obtained by inserting resistance in the secondary circuit and usually provides for a 50 per cent speed reduction when the motor takes its full rated current at normal speed. As resistors are supplied for both fan duty and constant torque duty, care should be taken in selecting the proper resistors.

Slip ring motors when used with centrifugal pumps and fans should have fan duty resistors. Because of the low current inrush of the fan and pump

load a starting resistor *NEMA* classification No. 15 may be used. For speed regulation resistor, classification No. 93 should be selected. On a compressor drive using an unloader, a constant torque resistor classification No. 15 should be used. If the compressor is started under load, *NEMA* classification No. 56 or 76 are used. For constant torque speed regulation, resistor No. 95 is used.

SINGLE PHASE MOTOR CONTROL

Where three phase current is not available or where single phase operation is preferred, then single phase repulsion induction, capacitor type or multispeed single phase motors may be used. Since the starting currents of all single phase motors are required to be within the starting-current limits established by the local power-supply company, a suitable type of starter may be chosen from the following selection:

1. Enclosed two pole manually operated motor starters with thermal overload protection.
2. Enclosed two pole automatic motor starter operated by a push button, thermostat or similar device, with thermal overload relay and low voltage protection.
3. A manual or magnetic resistance type starter with low voltage protection.
4. A manual or magnetic control for pole changing motors and for adjustable varying speed motors using an auto-transformer or resistance in the primary circuit to obtain line (or terminal) voltage drop.

In selecting across-the-line control for single phase capacitor type motors it is usually very desirable to use three pole across-the-line starters. Control for multispeed, single phase capacitor motors may be selected from tables on three phase rating when consideration is given to the increased current and the necessary switching of connections.

PROBLEMS IN PRACTICE

1 ● When motors are being considered as prime movers, what are some of the basic considerations that determine the final selection of the correct unit?

a. The kind of current available for driving the necessary motors is a primary consideration. There are two groups of motors available for driving the equipment on any job, which are the direct current type or alternating current type. The proper group selection depends entirely on the current available.

b It is also necessary to decide whether constant speed or variable speed operation is desired.

c. Consideration must also be given to the type of service required.

1. Whether variable torque or constant torque motors will be required.
2. Whether a high starting torque is required or whether a relatively small starting torque is required.

d. It is important to take into consideration the atmospheric conditions surrounding the motor location.

2 ● When using direct current motors: *a.* What three types are available as regards their windings; *b.* What four types are available with reference to their speed characteristics?

a. Shunt wound, compound wound, and series wound.

b. Constant speed, adjustable speed, adjustable varying speed, and varying speed.

3 ● With direct current motors as prime movers what type would you use: *a.* For driving a fan; *b.* For driving a compressor?

a. A fan requires a relatively small starting torque, therefore, a shunt wound motor would be ideal for this type service.

b. A compressor has a constant torque, therefore, a compound wound motor would be the proper selection for this duty.

4 ● What is one of the important factors that should be taken into account when a series wound direct current motor is being considered?

With a series wound motor the speed varies with the load, therefore this type should never be used where there is a possibility of the motor operating without being loaded. The resultant high speed may prove to be dangerous.

5 ● With the use of alternating current motors what two groups are generally considered?

Motors using single and polyphase power supply.

6 ● Under the alternating current group of motors what common types are available: *a.* For single phase duty; *b.* For polyphase duty?

a. Capacitor high torque, capacitor fan—(1) high torque, (2) low torque, capacitor start-induction run, repulsion induction, and split phase.

b. Squirrel cage—(1) general purpose, (2) medium torque, (3) high torque, automatic start high torque, normal torque normal current, normal torque low current, high torque low current, slip ring wound rotor, synchronous high speed, synchronous low speed.

7 ● What is the most commonly used of the polyphase motors?

The squirrel cage induction motor is the type most generally used for ordinary application.

8 ● With the use of squirrel cage motors what speed characteristic is available and what construction is used to make these more flexible?

The squirrel cage motor is basically a constant speed motor. However both single phase and polyphase high slip motors are used for adjustable varying speed drive through the use of line voltage control. When using an adjustable varying speed motor, particularly with a centrifugal fan and to a somewhat lesser extent, with a propeller fan, special care should be taken to assure that the fan is closely motored (*i.e.*, adequately loads the motor) in order to obtain the desired speeds under reduced speed operation. To make the squirrel cage motor more flexible, multispeed units are used quite frequently. These units may be single winding for the two speed unit or for different number of windings depending upon the number and combination of speeds required. For two speed single winding units the second speed is always one-half of top speed.

9 ● Differentiate between synchronous speed and full load speed of a motor.

Synchronous speed is the theoretical or no load speed. With the induction motor there is a certain amount of slip depending upon the load. As a rule, at full load, the speed is approximately 96 per cent of synchronous speed, however, motor manufacturers generally list full load speeds on their motor name plates.

The synchronous type of motor has a full load speed which is the same as the synchronous.

10 ● What are the general requirements usually recommended by the power company with reference to connecting polyphase motors to the power line?

For motors up to and including 5 hp, normal torque, normal starting current type of units can be connected directly to the line.

For motors from 5 to 30 hp, both high torque and normal torque, low starting current types of units can be used with across-the-line type of control.

Above these sizes, it is necessary to furnish current limiting starting equipment.

It is always advisable to check with local power companies as there are no standards for

connecting of loads on the power line and they are likely to vary with different power companies.

11 ● In controlling direct current motors what two methods are used, what speed ranges are obtained, and what is the relative efficiency of each method?

In controlling direct current motors, resistance is placed in either the armature circuit or the field circuit. For armature control, the speed is reduced with the increase of resistance. With the field control, the speed is increased with the addition of resistance in the field circuit.

For most listed direct current motors, it is possible to obtain operation up to a speed ratio of two to one with field control equipment. This type of control is used in connection with shunt wound motors for best results.

For speed adjustment by resistance in series with the armature circuit, a reduction of 50 per cent in speed can generally be obtained. This control can be used with either shunt or compound wound motors.

The field control method of changing speeds on direct current motors is the most efficient. Due to the large current in the armature circuit, this method results in a high loss when the speed is reduced any appreciable amount. It is well to remember that with field control only constant horsepower output is obtained, therefore, care should be taken that the motor at normal speed is large enough to care for any increase in load as a result of speeding up the unit.

12 ● What reduction in speed is possible and how is it obtained when alternating current slip ring motors are used?

Speed variation in slip ring motors is obtained by inserting resistance in the secondary circuit. This generally allows for a 50 per cent speed reduction when it is fully loaded at normal speed.

From 20 to 30 per cent speed reduction can be obtained through the use of line voltage control of an adjustable varying speed motor with a fan closely motored (*i.e.*, the fan approximately fully loads the motor).

Chapter 39

PIPING AND DUCT INSULATION

Heat Losses from Bare and Insulated Pipes, Heat Losses from Ducts, Low Temperature Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

INSULATION reduces the flow of heat where it is desired to maintain a temperature higher or lower than that of the surroundings. Its use contributes to the most economical operation of heating and refrigerating systems.

HEAT LOSSES FROM BARE PIPE

Heat losses from horizontal bare iron pipes, based on data obtained from tests conducted at the *Mellon Institute*, are given in Table 1. The

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE IRON PIPES

Expressed in Btu per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.543	0.573	0.605	0.638	0.656	0.742	0.796
¾	0.660	0.690	0.729	0.762	0.781	0.886	0.955
1	0.791	0.829	0.878	0.920	0.953	1.084	1.166
1¼	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1½	1.09	1.15	1.220	1.29	1.335	1.520	1.640
2	1.34	1.40	1.491	1.58	1.637	1.866	2.015
2½	1.58	1.67	1.778	1.87	1.937	2.215	2.388
3	1.88	1.99	2.100	2.22	2.301	2.641	2.853
3½	2.13	2.24	2.380	2.51	2.585	2.972	3.215
4	2.36	2.50	2.650	2.78	2.873	3.312	3.582
4½	2.60	2.75	2.920	3.08	3.170	3.655	3.956
5	2.87	3.02	3.200	3.38	3.493	4.030	4.368
6	3.39	3.56	3.775	4.01	4.115	4.755	5.153
8	4.32	4.55	4.830	5.14	5.270	6.120	6.635
10	5.32	5.61	5.925	6.34	6.551	7 592	8.245
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

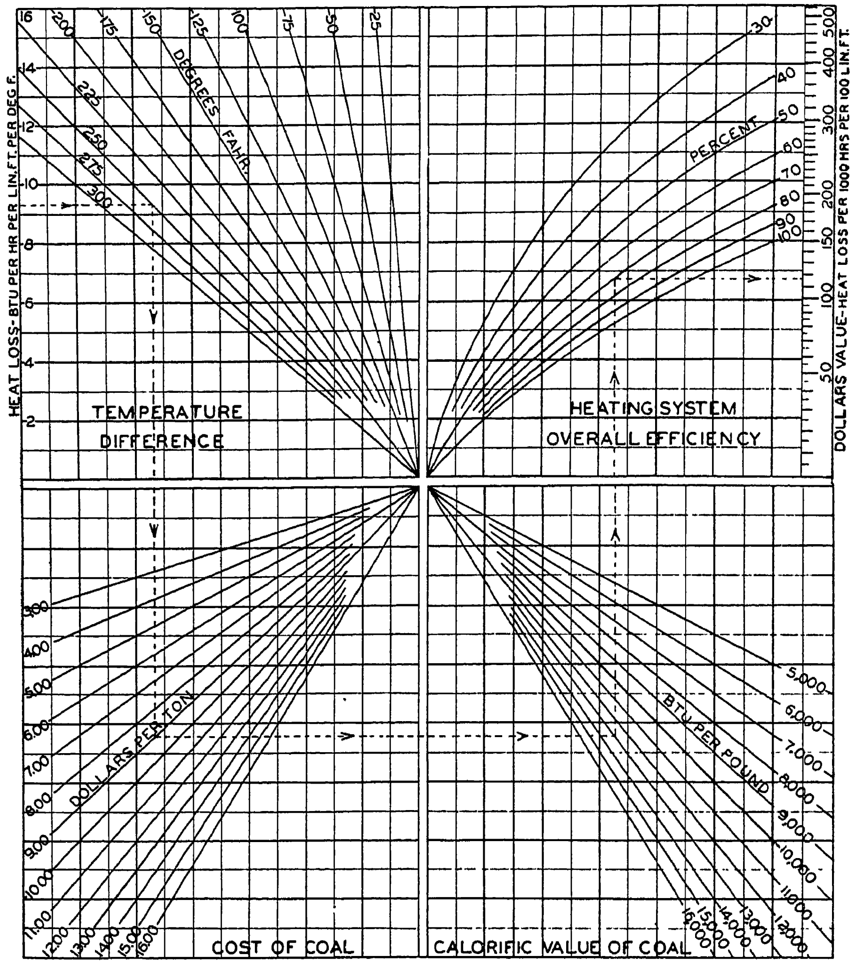


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 1)[a]

[a]This chart is based on 100 linear feet per 1000 hours. For fractions or multiples of these factors, multiply by proper percentage.

monetary value of the loss of heat given in Table 1 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 1 and locate this value on the upper left hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right hand scale. In using this chart, the cost of coal should also include the labor for handling it, boiler room expense, etc.

TABLE 2. HEAT LOSS FROM HORIZONTAL BARE BRIGHT COPPER PIPE

Expressed in Btu per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.180	0.210	0.218	0.229	0.299	0.338	0.355
¾	0.236	0.275	0.291	0.307	0.357	0.408	0.418
1	0.290	0.338	0.354	0.373	0.440	0.492	0.523
1¼	0.340	0.400	0.418	0.443	0.510	0.571	0.598
1½	0.390	0.463	0.473	0.507	0.598	0.671	0.710
2	0.490	0.525	0.600	0.628	0.719	0.813	0.851
2½	0.580	0.675	0.709	0.750	0.840	0.953	1.008
3	0.680	0.788	0.848	0.871	0.987	1.107	1.165
3½	0.760	0.888	0.946	1.000	1.114	1.235	1.307
4	0.940	1.000	1.045	1.107	1.210	1.361	1.456
4½	.....	.....	.....	.....	1.335	1.495	1.488
5	1.020	1.200	1.255	1.320	1.465	1.670	1.755
6	1.160	1.375	1.410	1.500	1.685	1.890	1.942
8	1.460	1.725	1.820	1.890	2.100	2.373	2.510

TABLE 3. HEAT LOSS FROM BRIGHT COPPER PIPE GIVEN ONE THIN COAT OF CLEAR LACQUER

Expressed in Btu per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.240	0.265	0.282	0.307	0.401	0.461	0.478
¾	0.320	0.356	0.373	0.414	0.477	0.571	0.578
1	0.390	0.437	0.463	0.507	0.598	0.681	0.710
1¼	0.470	0.537	0.554	0.614	0.700	0.812	0.840
1½	0.540	0.612	0.645	0.714	1.208	0.966	0.990
2	0.690	0.762	0.818	0.892	1.005	1.164	1.201
2½	0.840	0.937	0.991	1.085	1.178	1.361	1.420
3	0.960	1.025	1.135	1.270	1.400	1.625	1.700
3½	1.100	1.250	1.318	1.442	1.580	1.845	1.905
4	1.241	1.400	1.480	1.556	1.750	2.040	2.130
4½	.....	.....	.....	.....	1.910	2.240	2.350
5	1.480	1.685	1.790	1.965	2.130	2.415	2.610
6	1.700	1.936	2.052	2.272	2.450	2.810	2.990
8	2.200	2.500	2.630	2.854	3.120	3.425	3.730

Heat losses from horizontal copper tubes and pipes with bright, bright lacquered and tarnished surfaces are given in Tables 2, 3 and 4[1].

In order to determine heat losses per linear foot of pipe from known losses per square foot, it is necessary to know the area in square feet per

[1] Heat Loss from Copper Piping, by R. H. Heilman (*Heating Piping and Air Conditioning*, September, 1933, p. 458).

TABLE 4. HEAT LOSS FROM HORIZONTAL TARNISHED COPPER PIPE

Expressed in Btu per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube) 120 F	150 F	180 F	210 F	Steam (Standard Pipe Size Pipe) 227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.250	0.287	0.300	0.321	0.433	0.500	0.530
¾	0.340	0.381	0.409	0.429	0.533	0.543	0.654
1	0.440	0.475	0.509	0 536	0 636	0.746	0.803
1¼	0.500	0.559	0.618	0.622	0.764	0.878	0.934
1½	0.580	0.656	0.710	0.750	0.904	1.053	1.120
2	0.730	0 825	0.890	0.957	1.101	1.273	1.364
2½	0.880	1 000	1.091	1.143	1.305	1.490	1.605
3	1.040	1.175	1.272	1.343	1.560	1.800	1.940
3½	1.180	1.350	1.454	1.535	1.750	2.020	2.170
4	1.460	1.500	1.635	1.715	1.941	2.240	2.430
4½	....	.....	.....	.....	2.131	2.465	2.650
5	1.600	1.812	1.980	2.071	2.387	2.770	2.990
6	1.840	2.125	2.270	2.430	2.740	3.210	3.440
8	2.400	2.685	2.910	3.110	3.310	4.050	4.370

linear foot of pipe. Table 5 gives these areas for various standard pipe sizes, and Table 6 for copper tubing, while Table 7 gives the area in square feet for flanges and fittings for various standard pipe sizes.

Very often, when pipes are insulated, flanges and fittings are left bare due to the belief that the losses from these parts are not large. However, the fact that a pair of 8 in. standard flanges having an area of 2.41 sq ft

TABLE 5. RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

TABLE 6. RADIATING SURFACE PER LINEAR FOOT OF COPPER TUBING

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
½	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949	..	.....
1½	0.426	4	1.080	..	.....

TABLE 7. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting.

would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water pipes are given in Table 8. In this table the conductivities are given as functions of the mean temperatures or the mean of the inner and outer surface temperatures of the insulations. This method of stating conductivities makes it possible to readily calculate the heat loss through single or compound sections. It should be emphasized that the conductivities given in Table 8 for the various insulations are the average of

TABLE 8. CONDUCTIVITIES (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

Types of Insulating Materials	Mean Temperature				
	100 F	200 F	300 F	400 F	500 F
85 per cent Magnesia Type	0.425	0.465	0.505	0.550	0.590
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0.530	0.650	0.770	0.890	
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0.480	0.555	0.630	0.705	
Laminated Asbestos Type (30–40 Laminations per 1 in. thick)	0.360	0.415	0.470	0.525	0.585
Laminated Asbestos Type (20 Laminations per 1 in. thick)	0.545	0.605	0.665	0.725	0.785
Rock Wool Type	0.350	0.410	0.470	0.530	0.590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.515	0.545	0.575	0.605	0.635
Brown Asbestos Type (Felted Fibre)	0.600	0.640	0.675	0.715	0.750

[a]R. H. Heilman, *Mechanical Engineering*, Vol. 46 (1924), p. 593.

TABLE 9. COEFFICIENTS OF TRANSMISSION (*U*) FOR PIPES INSULATED WITH 85 PER CENT MAGNESIA TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.744	0.754	0.764	0.774	0.779	0.802	0.814
	¾	0.672	0.681	0.689	0.697	0.701	0.721	0.731
	1	0.613	0.621	0.629	0.637	0.641	0.659	0.670
	1¼	0.562	0.570	0.577	0.585	0.589	0.606	0.617
	1½	0.532	0.539	0.546	0.553	0.557	0.573	0.582
	2	0.500	0.506	0.512	0.519	0.523	0.538	0.547
	2½	0.475	0.481	0.487	0.493	0.497	0.512	0.520
	3	0.455	0.461	0.467	0.474	0.477	0.492	0.500
1	3½	0.441	0.447	0.452	0.458	0.462	0.475	0.483
	4	0.429	0.435	0.441	0.446	0.449	0.463	0.471
	4½	0.420	0.425	0.431	0.437	0.440	0.453	0.460
	5	0.411	0.416	0.422	0.427	0.430	0.443	0.450
	6	0.402	0.408	0.413	0.419	0.422	0.435	0.442
	8	0.387	0.392	0.397	0.403	0.405	0.418	0.425
	10	0.375	0.380	0.385	0.390	0.393	0.405	0.412
	12	0.369	0.374	0.378	0.383	0.386	0.398	0.405
	½	0.617	0.625	0.633	0.642	0.646	0.665	0.676
	¾	0.550	0.558	0.566	0.573	0.577	0.596	0.606
	1	0.496	0.503	0.511	0.518	0.522	0.540	0.549
	1¼	0.453	0.459	0.465	0.472	0.475	0.490	0.498
	1½	0.424	0.430	0.436	0.442	0.445	0.459	0.467
	2	0.394	0.400	0.405	0.410	0.413	0.427	0.434
	2½	0.371	0.376	0.382	0.386	0.389	0.401	0.408
	3	0.352	0.357	0.362	0.367	0.370	0.380	0.387
1½	3½	0.339	0.343	0.347	0.351	0.354	0.364	0.370
	4	0.328	0.333	0.337	0.341	0.343	0.353	0.359
	4½	0.320	0.324	0.328	0.332	0.334	0.343	0.350
	5	0.312	0.316	0.320	0.324	0.326	0.336	0.342
	6	0.303	0.307	0.311	0.315	0.318	0.328	0.333
	8	0.287	0.291	0.295	0.299	0.301	0.311	0.316
	10	0.276	0.280	0.284	0.288	0.290	0.299	0.304
	12	0.272	0.275	0.279	0.283	0.285	0.294	0.299
	½	0.543	0.551	0.558	0.565	0.569	0.587	0.597
	¾	0.484	0.490	0.497	0.503	0.507	0.523	0.532
	1	0.433	0.439	0.445	0.451	0.454	0.467	0.476
	1¼	0.393	0.398	0.403	0.409	0.412	0.424	0.432
	1½	0.365	0.370	0.376	0.381	0.384	0.397	0.402
	2	0.338	0.343	0.347	0.351	0.354	0.364	0.370
	2½	0.316	0.320	0.324	0.328	0.331	0.341	0.347
	3	0.297	0.301	0.305	0.309	0.312	0.321	0.326
2	3½	0.284	0.288	0.292	0.295	0.297	0.306	0.311
	4	0.275	0.278	0.282	0.285	0.287	0.296	0.301
	4½	0.266	0.270	0.273	0.276	0.278	0.286	0.290
	5	0.258	0.262	0.265	0.268	0.270	0.278	0.283
	6	0.250	0.254	0.257	0.260	0.262	0.270	0.274
	8	0.236	0.239	0.242	0.245	0.247	0.255	0.258
	10	0.224	0.227	0.230	0.233	0.235	0.242	0.246
	12	0.219	0.222	0.225	0.228	0.230	0.237	0.240

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (4 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.890	0.919	0.949	0.978	0.995	1.065	1.106
	¾	0.803	0.829	0.857	0.883	0.898	0.961	0.997
	1	0.731	0.756	0.780	0.804	0.818	0.876	0.909
	1¼	0.671	0.693	0.716	0.738	0.751	0.804	0.834
	1½	0.635	0.656	0.677	0.698	0.710	0.760	0.788
	2	0.595	0.615	0.635	0.656	0.667	0.715	0.742
	2½	0.567	0.586	0.605	0.624	0.635	0.680	0.705
	3	0.544	0.562	0.580	0.598	0.608	0.652	0.677
	3½	0.527	0.544	0.561	0.578	0.588	0.631	0.654
	4	0.513	0.530	0.548	0.565	0.575	0.616	0.639
	4½	0.502	0.518	0.535	0.551	0.561	0.601	0.624
	5	0.490	0.507	0.523	0.539	0.549	0.588	0.611
	6	0.480	0.496	0.512	0.528	0.538	0.577	0.599
	8	0.462	0.477	0.493	0.508	0.517	0.554	0.575
	10	0.447	0.462	0.476	0.491	0.500	0.537	0.557
	12	0.441	0.456	0.470	0.485	0.493	0.529	0.550
1½	½	0.737	0.762	0.787	0.812	0.826	0.884	0.918
	¾	0.657	0.679	0.702	0.725	0.737	0.790	0.820
	1	0.594	0.614	0.634	0.654	0.666	0.713	0.740
	1¼	0.542	0.559	0.577	0.596	0.606	0.649	0.673
	1½	0.507	0.524	0.541	0.558	0.568	0.609	0.632
	2	0.471	0.487	0.503	0.519	0.528	0.565	0.587
	2½	0.443	0.458	0.473	0.488	0.497	0.533	0.553
	3	0.421	0.435	0.449	0.463	0.472	0.506	0.525
	3½	0.403	0.417	0.430	0.443	0.451	0.483	0.502
	4	0.393	0.405	0.418	0.432	0.439	0.471	0.489
	4½	0.383	0.394	0.407	0.420	0.428	0.460	0.476
	5	0.372	0.384	0.397	0.409	0.417	0.447	0.463
	6	0.362	0.374	0.387	0.399	0.406	0.436	0.452
	8	0.343	0.354	0.366	0.378	0.385	0.413	0.429
	10	0.328	0.339	0.351	0.362	0.369	0.397	0.413
	12	0.323	0.334	0.346	0.357	0.364	0.391	0.407
2	½	0.648	0.670	0.692	0.713	0.726	0.779	0.810
	¾	0.578	0.598	0.617	0.637	0.648	0.694	0.720
	1	0.518	0.535	0.552	0.570	0.580	0.622	0.645
	1¼	0.469	0.485	0.501	0.517	0.527	0.566	0.587
	1½	0.438	0.452	0.467	0.481	0.490	0.526	0.545
	2	0.404	0.417	0.430	0.444	0.452	0.483	0.502
	2½	0.379	0.391	0.403	0.415	0.422	0.451	0.466
	3	0.356	0.367	0.378	0.390	0.397	0.425	0.440
	3½	0.339	0.350	0.361	0.373	0.380	0.406	0.421
	4	0.328	0.339	0.350	0.360	0.367	0.392	0.406
	4½	0.318	0.328	0.339	0.350	0.357	0.381	0.395
	5	0.308	0.318	0.329	0.340	0.346	0.370	0.384
	6	0.299	0.309	0.319	0.329	0.335	0.358	0.371
	8	0.282	0.291	0.301	0.310	0.315	0.336	0.349
	10	0.267	0.276	0.285	0.294	0.299	0.319	0.332
	12	0.263	0.272	0.280	0.289	0.294	0.314	0.325

TABLE 11. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (8 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.801	0.820	0.838	0.857	0.868	0.913	0.939
	¾	0.723	0.739	0.756	0.773	0.783	0.824	0.847
	1	0.658	0.673	0.688	0.704	0.713	0.751	0.772
	1¼	0.606	0.619	0.633	0.647	0.655	0.688	0.707
	1½	0.573	0.586	0.599	0.612	0.619	0.652	0.670
	2	0.538	0.550	0.562	0.575	0.581	0.612	0.629
	2½	0.511	0.523	0.534	0.546	0.553	0.582	0.599
	3	0.489	0.501	0.512	0.524	0.531	0.558	0.575
	3½	0.474	0.485	0.496	0.507	0.514	0.542	0.557
	4	0.461	0.472	0.482	0.493	0.500	0.527	0.542
	4½	0.451	0.462	0.472	0.482	0.489	0.515	0.530
	5	0.442	0.452	0.462	0.473	0.479	0.505	0.520
	6	0.432	0.442	0.452	0.463	0.468	0.493	0.508
	8	0.416	0.426	0.436	0.446	0.451	0.475	0.489
	10	0.402	0.412	0.421	0.430	0.435	0.459	0.473
	12	0.397	0.406	0.415	0.424	0.429	0.452	0.466
1½	½	0.664	0.679	0.695	0.711	0.720	0.759	0.780
	¾	0.593	0.607	0.621	0.636	0.643	0.677	0.697
	1	0.535	0.547	0.560	0.573	0.580	0.611	0.629
	1¼	0.488	0.499	0.510	0.522	0.528	0.556	0.572
	1½	0.457	0.467	0.478	0.490	0.496	0.522	0.537
	2	0.425	0.434	0.444	0.455	0.460	0.485	0.499
	2½	0.399	0.408	0.418	0.428	0.434	0.457	0.471
	3	0.378	0.387	0.396	0.405	0.411	0.433	0.446
	3½	0.363	0.371	0.380	0.388	0.393	0.415	0.427
	4	0.353	0.361	0.369	0.378	0.383	0.403	0.415
	4½	0.343	0.351	0.360	0.368	0.373	0.393	0.404
	5	0.334	0.342	0.350	0.358	0.363	0.383	0.394
	6	0.325	0.333	0.341	0.349	0.353	0.373	0.383
	8	0.309	0.316	0.324	0.332	0.336	0.355	0.365
	10	0.295	0.303	0.310	0.318	0.322	0.340	0.350
	12	0.291	0.298	0.306	0.313	0.317	0.335	0.344
2	½	0.585	0.599	0.613	0.627	0.635	0.668	0.688
	¾	0.520	0.533	0.545	0.558	0.565	0.595	0.612
	1	0.465	0.476	0.487	0.498	0.504	0.532	0.547
	1¼	0.422	0.432	0.442	0.452	0.458	0.483	0.497
	1½	0.394	0.403	0.412	0.422	0.427	0.450	0.462
	2	0.364	0.372	0.380	0.388	0.393	0.415	0.427
	2½	0.339	0.347	0.355	0.363	0.367	0.387	0.398
	3	0.319	0.327	0.334	0.342	0.346	0.365	0.375
	3½	0.304	0.311	0.318	0.326	0.330	0.349	0.358
	4	0.295	0.302	0.308	0.315	0.319	0.336	0.345
	4½	0.285	0.292	0.299	0.306	0.310	0.327	0.336
	5	0.278	0.284	0.290	0.297	0.301	0.317	0.326
	6	0.269	0.275	0.282	0.288	0.292	0.307	0.315
	8	0.253	0.259	0.265	0.270	0.273	0.288	0.296
	10	0.240	0.245	0.251	0.257	0.260	0.275	0.282
	12	0.236	0.241	0.247	0.253	0.256	0.270	0.277

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (30 TO 40 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.605	0.620	0.635	0.650	0.658	0.695	0.716
	¾	0.546	0.560	0.573	0.586	0.594	0.627	0.645
	1	0.498	0.510	0.522	0.534	0.541	0.570	0.587
	1¼	0.457	0.468	0.480	0.491	0.497	0.525	0.540
	1½	0.432	0.442	0.453	0.464	0.470	0.496	0.511
	2	0.406	0.416	0.426	0.437	0.442	0.467	0.481
	2½	0.385	0.395	0.405	0.415	0.420	0.443	0.457
	3	0.370	0.379	0.389	0.398	0.403	0.425	0.438
	3½	0.359	0.367	0.376	0.385	0.390	0.413	0.426
	4	0.349	0.358	0.366	0.375	0.380	0.402	0.414
	4½	0.341	0.350	0.359	0.367	0.372	0.393	0.405
	5	0.334	0.342	0.351	0.359	0.364	0.384	0.395
	6	0.327	0.335	0.343	0.351	0.356	0.376	0.387
	8	0.314	0.322	0.330	0.338	0.343	0.362	0.373
	10	0.304	0.312	0.320	0.328	0.332	0.350	0.361
	12	0.301	0.308	0.316	0.324	0.328	0.346	0.356
1½	½	0.502	0.514	0.526	0.539	0.546	0.577	0.595
	¾	0.450	0.461	0.473	0.484	0.490	0.517	0.532
	1	0.405	0.415	0.426	0.436	0.442	0.466	0.480
	1¼	0.369	0.378	0.387	0.396	0.401	0.423	0.435
	1½	0.343	0.352	0.361	0.370	0.375	0.397	0.409
	2	0.321	0.329	0.337	0.345	0.350	0.369	0.380
	2½	0.301	0.309	0.317	0.324	0.330	0.348	0.358
	3	0.286	0.293	0.301	0.308	0.313	0.330	0.340
	3½	0.274	0.281	0.288	0.295	0.300	0.316	0.326
	4	0.267	0.273	0.280	0.287	0.291	0.307	0.317
	4½	0.259	0.266	0.272	0.279	0.283	0.299	0.308
	5	0.253	0.260	0.266	0.272	0.276	0.291	0.300
	6	0.247	0.253	0.260	0.266	0.269	0.284	0.293
	8	0.234	0.240	0.246	0.252	0.255	0.270	0.279
	10	0.223	0.229	0.235	0.241	0.245	0.258	0.266
	12	0.221	0.227	0.232	0.238	0.241	0.255	0.263
2	½	0.442	0.453	0.464	0.475	0.481	0.508	0.523
	¾	0.392	0.402	0.412	0.422	0.428	0.452	0.465
	1	0.352	0.360	0.369	0.378	0.383	0.405	0.417
	1¼	0.319	0.327	0.335	0.343	0.348	0.367	0.379
	1½	0.297	0.304	0.311	0.319	0.323	0.341	0.352
	2	0.274	0.280	0.287	0.294	0.298	0.314	0.324
	2½	0.256	0.262	0.269	0.275	0.279	0.293	0.302
	3	0.243	0.249	0.254	0.260	0.264	0.277	0.285
	3½	0.231	0.236	0.242	0.248	0.251	0.265	0.273
	4	0.223	0.228	0.234	0.240	0.243	0.257	0.265
	4½	0.216	0.222	0.227	0.233	0.236	0.249	0.256
	5	0.210	0.215	0.220	0.225	0.228	0.241	0.248
	6	0.203	0.208	0.213	0.218	0.221	0.233	0.240
	8	0.191	0.196	0.201	0.206	0.209	0.220	0.227
	10	0.182	0.187	0.192	0.196	0.199	0.210	0.215
	12	0.178	0.183	0.187	0.192	0.195	0.205	0.210

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (APPROXIMATELY 20 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.910	0.925	0.940	0.956	0.964	1.001	1.022
	¾	0.823	0.836	0.850	0.863	0.871	0.902	0.921
	1	0.748	0.760	0.773	0.785	0.792	0.823	0.840
	1¼	0.686	0.698	0.710	0.721	0.728	0.756	0.771
	1½	0.649	0.659	0.671	0.682	0.688	0.716	0.731
	2	0.610	0.620	0.630	0.640	0.647	0.671	0.685
	2½	0.581	0.590	0.600	0.609	0.615	0.638	0.651
	3	0.558	0.567	0.576	0.585	0.591	0.613	0.626
	3½	0.539	0.548	0.557	0.566	0.571	0.592	0.604
	4	0.524	0.532	0.541	0.551	0.556	0.577	0.589
	4½	0.514	0.522	0.530	0.539	0.544	0.564	0.575
	5	0.503	0.511	0.519	0.528	0.533	0.553	0.565
	6	0.492	0.500	0.509	0.517	0.522	0.542	0.553
	8	0.473	0.480	0.488	0.497	0.502	0.521	0.532
	10	0.458	0.465	0.473	0.481	0.485	0.504	0.514
	12	0.452	0.459	0.467	0.475	0.478	0.497	0.507
1½	½	0.755	0.767	0.780	0.793	0.800	0.831	0.848
	¾	0.674	0.685	0.697	0.708	0.715	0.743	0.759
	1	0.607	0.618	0.628	0.639	0.645	0.670	0.684
	1¼	0.553	0.562	0.572	0.581	0.587	0.610	0.622
	1½	0.517	0.527	0.536	0.545	0.550	0.572	0.584
	2	0.481	0.490	0.499	0.508	0.513	0.535	0.547
	2½	0.453	0.460	0.469	0.477	0.481	0.500	0.511
	3	0.429	0.436	0.444	0.452	0.456	0.475	0.485
	3½	0.412	0.419	0.427	0.434	0.438	0.456	0.465
	4	0.400	0.407	0.415	0.422	0.426	0.443	0.453
	4½	0.390	0.396	0.402	0.409	0.413	0.429	0.437
	5	0.380	0.386	0.393	0.400	0.403	0.418	0.427
	6	0.369	0.375	0.382	0.389	0.392	0.408	0.417
	8	0.351	0.358	0.364	0.370	0.374	0.388	0.397
	10	0.337	0.344	0.350	0.356	0.359	0.373	0.382
	12	0.332	0.338	0.344	0.350	0.353	0.367	0.375
2	½	0.664	0.675	0.687	0.698	0.704	0.732	0.747
	¾	0.591	0.601	0.611	0.621	0.627	0.652	0.665
	1	0.529	0.538	0.547	0.557	0.562	0.584	0.597
	1¼	0.480	0.488	0.497	0.505	0.510	0.529	0.540
	1½	0.445	0.453	0.462	0.470	0.475	0.494	0.504
	2	0.412	0.420	0.427	0.434	0.438	0.455	0.464
	2½	0.385	0.392	0.398	0.405	0.409	0.425	0.434
	3	0.364	0.370	0.376	0.382	0.385	0.400	0.408
	3½	0.346	0.352	0.358	0.365	0.368	0.382	0.390
	4	0.336	0.342	0.348	0.354	0.357	0.371	0.378
	4½	0.325	0.332	0.338	0.343	0.346	0.360	0.367
	5	0.316	0.322	0.327	0.333	0.336	0.349	0.356
	6	0.306	0.312	0.317	0.323	0.326	0.338	0.345
	8	0.288	0.293	0.298	0.303	0.306	0.317	0.324
	10	0.275	0.279	0.284	0.289	0.292	0.302	0.308
	12	0.269	0.274	0.278	0.283	0.286	0.296	0.302

TABLE 14. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH ROCK WOOL TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.631	0.644	0.658	0.672	0.680	0.712	0.730
	¾	0.569	0.581	0.593	0.606	0.613	0.642	0.659
	1	0.518	0.529	0.541	0.552	0.559	0.585	0.600
	1¼	0.476	0.486	0.497	0.507	0.513	0.537	0.551
	1½	0.450	0.460	0.470	0.480	0.485	0.508	0.522
	2	0.422	0.431	0.441	0.450	0.456	0.478	0.490
	2½	0.402	0.411	0.420	0.428	0.434	0.455	0.466
	3	0.385	0.394	0.402	0.411	0.415	0.435	0.446
1	3½	0.373	0.381	0.389	0.398	0.402	0.421	0.432
	4	0.363	0.371	0.379	0.387	0.392	0.411	0.422
	4½	0.355	0.363	0.371	0.379	0.383	0.402	0.413
	5	0.348	0.356	0.364	0.371	0.376	0.394	0.404
	6	0.341	0.348	0.356	0.363	0.368	0.386	0.396
	8	0.327	0.335	0.342	0.349	0.353	0.372	0.381
	10	0.317	0.324	0.331	0.338	0.343	0.360	0.369
	12	0.313	0.320	0.327	0.334	0.338	0.355	0.364
	½	0.523	0.534	0.545	0.556	0.563	0.590	0.606
	¾	0.468	0.477	0.487	0.497	0.503	0.528	0.542
	1	0.421	0.430	0.440	0.449	0.455	0.477	0.490
	1¼	0.383	0.391	0.399	0.407	0.412	0.433	0.444
	1½	0.359	0.366	0.375	0.383	0.387	0.407	0.419
	2	0.333	0.340	0.348	0.356	0.360	0.378	0.389
	2½	0.314	0.320	0.327	0.335	0.339	0.355	0.365
	3	0.296	0.302	0.310	0.317	0.321	0.337	0.347
1½	3½	0.286	0.291	0.298	0.304	0.307	0.323	0.332
	4	0.278	0.284	0.290	0.296	0.300	0.315	0.323
	4½	0.270	0.276	0.282	0.287	0.291	0.305	0.313
	5	0.263	0.269	0.275	0.280	0.284	0.298	0.305
	6	0.257	0.262	0.267	0.273	0.277	0.290	0.297
	8	0.244	0.249	0.254	0.260	0.263	0.276	0.283
	10	0.235	0.240	0.245	0.250	0.253	0.265	0.272
	12	0.230	0.234	0.239	0.245	0.247	0.260	0.267
	½	0.461	0.471	0.481	0.491	0.496	0.520	0.534
	¾	0.409	0.418	0.427	0.436	0.441	0.463	0.475
	1	0.366	0.374	0.382	0.390	0.395	0.415	0.427
	1¼	0.333	0.340	0.347	0.355	0.359	0.377	0.387
	1½	0.310	0.316	0.323	0.330	0.334	0.351	0.360
	2	0.286	0.292	0.298	0.304	0.308	0.323	0.331
	2½	0.268	0.274	0.279	0.285	0.289	0.302	0.310
	3	0.252	0.257	0.262	0.268	0.272	0.284	0.292
2	3½	0.241	0.246	0.251	0.257	0.260	0.272	0.280
	4	0.232	0.237	0.242	0.247	0.250	0.262	0.269
	4½	0.225	0.230	0.235	0.240	0.243	0.255	0.262
	5	0.218	0.223	0.228	0.233	0.236	0.247	0.253
	6	0.213	0.217	0.221	0.226	0.228	0.239	0.245
	8	0.200	0.204	0.208	0.213	0.215	0.225	0.231
	10	0.189	0.193	0.197	0.201	0.204	0.214	0.220
	12	0.185	0.190	0.194	0.198	0.200	0.210	0.216

values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials, will of course, vary in conductivity to some extent from these values.

The heat losses through six of the types of insulation given in Table 8 for 1, 1½ and 2 in. thick materials, and for temperatures commonly encountered in engineering practice can be obtained from Tables 9 to 14 inclusive. The loss through other thicknesses of the materials, and for other hot water or steam temperature conditions may be obtained by interpolation. The heat loss coefficients given in Tables 9 to 14 are based on the conductivities in Table 8 and were computed from data given in Chapter 22, THE GUIDE 1931.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces the increases in losses due to air velocity are very small as compared with increases shown for bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in loss due to air velocity ranges from about 30 per cent in the case of 1 in. thick insulation, to about 10 per cent in the case of 3 in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as possible. Pipe insulation out-of-doors should be provided with a waterproof jacket, and other outdoor insulation should be thoroughly weatherproofed.

HEAT LOSSES FROM DUCTS

The heat transmission through sheet metal duct walls is mainly a function of the surface character of the metal since the thickness of the metal itself is not enough to appreciably retard the flow of heat. In other words, the two surfaces provide the resistance to heat flow through the metal. The surfaces of black iron probably offer the least resistance to the flow of heat, while metals with brighter and smoother surfaces, offer greater resistance. For ducts in service at normal air velocities and temperatures, the coefficient of heat transmission for black iron is 1.6 Btu per square foot per hour per degree Fahrenheit difference between the mean temperature in the duct and the temperature of the surrounding air and for galvanized iron is 1.1 Btu per square foot per hour per degree Fahrenheit difference.

The heat loss from a given length of duct is expressed by:

$$H = k\,P\,L\left[\left(\frac{t_1 + t_2}{2}\right) - t_s\right] \qquad (1)$$

The heat given up by the air in the duct is:

$$H = 0.24\,M\,(t_1 - t_2) = 14.4\,A\,V\,d\,(t_1 - t_2) \qquad (2)$$

Equating 1 and 2 enables the determination of the temperature drop in the duct:

$$k\,P\,L\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] = 14.4\,A\,V\,d\,(t_1 - t_2)$$

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8\,A\,V\,d}{k\,P\,L} \tag{3}$$

where

H = heat loss through duct walls, Btu per hour.
k = overall heat transmission coefficient, Btu per square foot per hour per degree Fahrenheit temperature difference.
P = perimeter of duct, feet.
L = length of duct, feet.
t_1 = temperature of air entering duct, degrees Fahrenheit.
t_2 = temperature of air leaving duct, degrees Fahrenheit.
t_3 = temperature of air surrounding duct, degrees Fahrenheit.
M = weight of air per hour through the duct, pounds.
A = cross-sectional area of duct, square feet.
V = velocity of air in the duct, feet per minute, at specified temperature.
d = density of air at the specified temperature at which V is measured.

In using the formula 3 one of the duct air temperatures will be unknown and will be solved for by substitution of the other known or assumed values. The assumed values dependent upon the mean duct air temperature can be determined exactly by cut and try.

Heat losses for insulated ducts are given in the warm air column of Table 15. The losses are based on a uniform series of material conductivities at 86 F mean temperature and an air temperature of 50 F outside of the duct. The losses may be interpolated for odd material conductivities and temperatures. The conductivities of various materials will be found in Table 2 of Chapter 5. For cases where the surrounding air temperature is other than 50 F the losses may be selected on the basis of temperature difference.

Example 1. Determine the entering air temperature and heat loss for a duct 24 x 36 in. cross section and 70 ft in length, insulated with ½ in. of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution. Assume the entering air temperature to be 130 F. Thus the mean temperature difference will be 85 F. Referring to the warm air column of Table 15 and interpolating for 90 F temperature difference, the overall heat transmission coefficient is found to be 0.516 Btu. From Table 4, Chapter 1 the density of air at 70 F and 29.92 in. Hg. is found to be 0.07423 lb per cubic foot. Substituting these and the other given values in Formula 3,

$$\frac{t_1 + 120 - (2 \times 40)}{t_1 - 120} = \frac{28.8 \times 6 \times 1200 \times 0.07423}{0.516 \times 10 \times 70}$$

$$t_1 + 120 - 80 = 42.62\,(t_1 - 120)$$

$$t_1 + 40 = 42.62\,t_1 - 5114$$

$$5154 = 41.62t_1$$

$$123.8 = t_1$$

Based on 123.8 F entering air temperature the new mean temperature difference will be

81.9 F and the new transmission coefficient will be 0.515. Resubstituting in Formula 3, t_1 becomes 123.9 F, which value is evidently exact within one tenth of one degree.

Substituting in Formula 1,

$$H = 0.515 \times 10 \times 70 \left[\left(\frac{123.9 + 120}{2} \right) - 40 \right]$$

$$H = 0.515 \times 10 \times 70 \times 81.95$$

$$H = 29{,}543 \text{ Btu.}$$

TABLE 15. HEAT TRANSMISSION THROUGH DUCT WALLS INSULATED WITH MATERIALS OF VARYING CONDUCTIVITIES[a]

Values are expressed in Btu per hour per square foot of flat surface per degree Fahrenheit difference in temperature between air inside and still air outside at 90 F for cold air and 50 F for warm air in ducts

Conductivity of Insulation at 86 F Mean Temp.	Thickness of Insulation (Inches)	Cold Air 40 F	Cold Air 60 F	Cold Air 80 F	Warm Air 90 F	Warm Air 120 F	Warm Air 150 F	Warm Air 180 F
		Temperature Difference 50 F	30 F	10 F	40 F	70 F	100 F	130 F
0 200	½	0.319	0.323	0.328	0.324	0.330	0.337	0.344
	1	0.175	0.177	0.180	0.178	0.181	0.184	0.188
	1½	0.121	0.122	0.124	.....	0.125	0.127	0.129
	2	0.092	0.093	0.095	. . .	.....	.....	.....
0.250	½	0.382	0.387	0.392	0.390	0.397	0.404	0 412
	1	0.214	0.217	0.220	0.218	0 221	0.225	0.229
	1½	0.149	0.151	0.153	.....	0.154	0.156	0.159
	2	0.114	0.115	0.117	....	.....	.....	.....
0.300	½	0.440	0.445	0.450	0.448	0.457	0.466	0.475
	1	0.252	0.255	0.258	0.256	0.260	0.264	0.268
	1½	0.176	0.178	0.180	.....	0.181	0.184	0.187
	2	0.135	0.137	0.139	.. ..	.....	.....	.....
0.350	½	0.494	0.499	0.505	0.502	0.511	0.521	0.530
	1	0.286	0.289	0.292	0.290	0.295	0.300	0.306
	1½	0.202	0.204	0.207	.. ..	0.208	0.211	0.215
	2	0.156	0.158	0.160	.....	.....	.....	.. ..
0.450	½	.....	0.596	0.602	0.599	0.610	0 621	0.633
	1	.....	0.356	0.360	0.358	0.364	0.370	0.376
	1½	.....	0.254	0.257	.....	0.259	0.263	0.267
	2	.....	0.198	0.200	....	.....	.. ..	.....
0.550	½	.....	0.682	0 688	0.685	0.699	0.714	0.730
	1	....	0.417	0.422	0.418	0.425	0.432	0.440
	1½	.....	0 302	0.305	. .	0.307	0.312	0.317
	2	....	0.236	0.239	...	.....	.....	....

[a]For round ducts less than 30 in. diameter, increase heat transmission values by the following percentages:

THICKNESS OF INSULATION (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

LOW TEMPERATURE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation and frost. The insulating material should absorb a minimum amount of moisture, for one reason that the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the case of surfaces to be insulated that are below the dew point of the surrounding air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface.

The thickness of insulation which should be used to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 2. The maximum permissable temperature drop is indicated at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop represents the difference between the dry-bulb temperature and the dew point temperature for the conditions involved. (See discussion of condensation in Chapter 7). The surface resistances used for calculating the family of curves in Fig. 2 are based on tests made on canvas covered pipe insulation surfaces at *Mellon Institute*. However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves may be followed with no alteration for surfaces commonly used.

Heat gains for pipes insulated with a material having a conductivity of 0.30 Btu per square foot per hour per degree Fahrenheit difference per inch thickness are given in Table 16.

Heat gains for insulated ducts are given in the cold air column of Table 15. The heat gains are based on a uniform series of conductivities at 86 F mean temperature and an air temperature of 90 F outside of the duct. The gains may be interpolated for odd material conductivities and temperatures. For cases where the surrounding air temperature is other than 90 F the gains may be selected on the basis of temperature difference.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 17 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and

surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 F above, and the surrounding air tem-

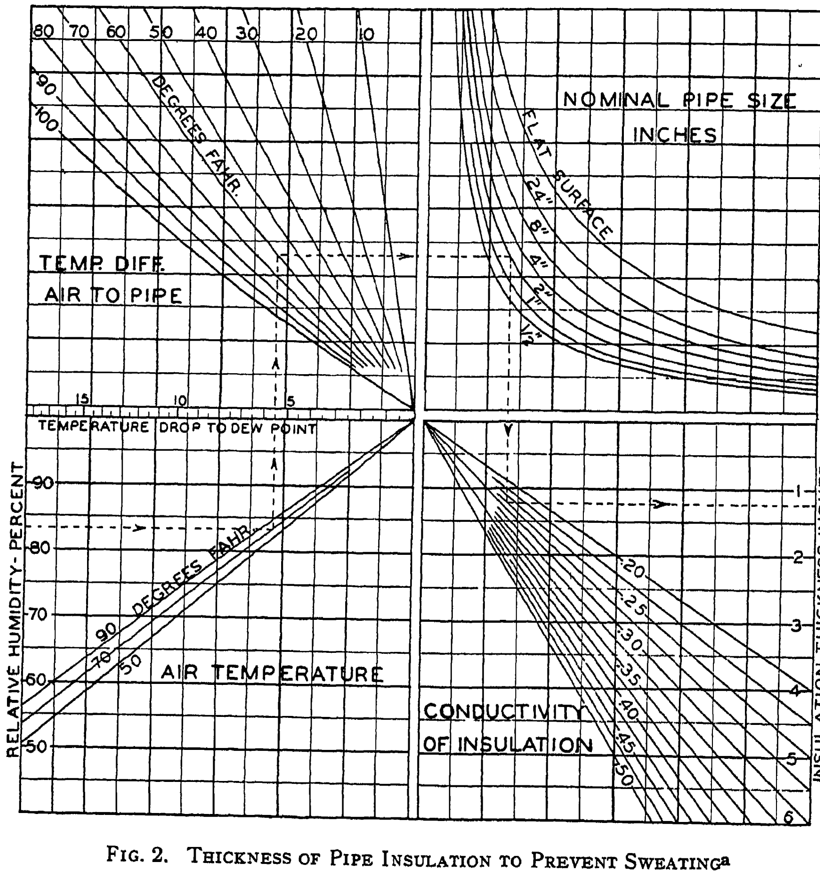


FIG. 2. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

perature 50 F below the freezing point of water (temperature difference, 60 F).

The last column of Table 17 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is ad-

visable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 17. However, if the water enters the pipe at 34 F it will be cooled to 32 F in

TABLE 16. HEAT GAINS FOR INSULATED COLD PIPES

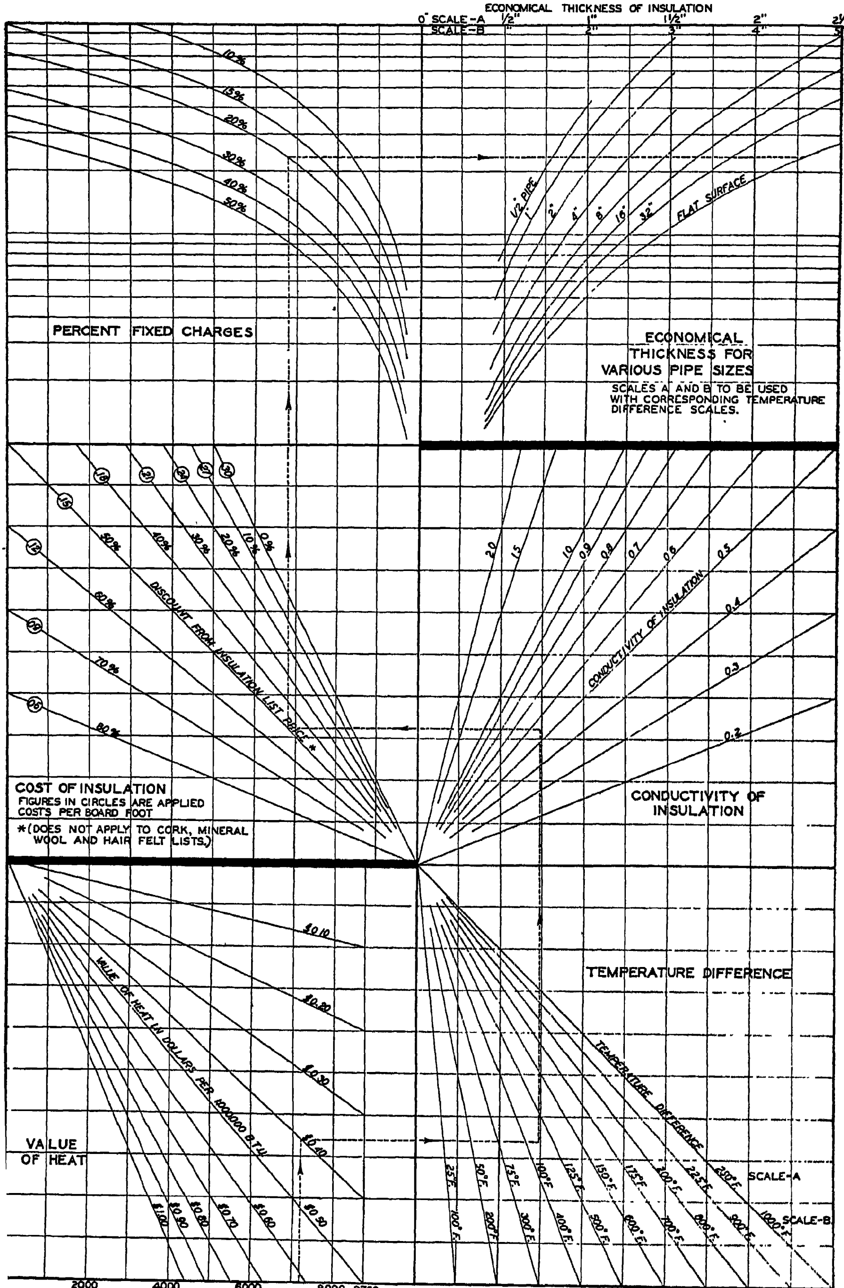
Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
1/2	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
3/4	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1 1/4	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1 1/2	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2 1/2	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3 1/2	1.5	0.295	0.282	2.9	0.191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0.383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 17, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.



(L. B. McMillan, *Proc. National Dist. Heating Ass'n.*, Vol. 18, p. 138.)

FIG. 3. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION

When water must remain stationary longer than the times designated in Table 17, the only safe way to insure against freezing is to install a steam or hot water line or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 18. Where a thorough analysis of economic thickness is desired this may be accomplished through the the use of the chart, Fig. 3.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 42). Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult of accurate determination due to the many variables which have to be considered. As a result of theories[3] previously developed, together

[2]Handbook of the *National District Heating Association*, Second Edition, 1932.

[3]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A. S. H. V. E. TRANSACTIONS Vol. 26, 1920, p. 335).

TABLE 17. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES

Nominal Pipe Size (Inches)	Number of Hours to Cool Water to Freezing Point			Water Required to Flow to Prevent Freezing Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

TABLE 18. THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

TABLE 19. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

with other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 19 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 3. The data in Fig. 3 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

PROBLEMS IN PRACTICE

1 ● What precautions must be taken in selecting insulation used for covering pipe lines carrying materials at temperatures lower than the dew point?

Materials intended for this service should be as moisture proof as possible and in addition an outer covering should be applied which is proof against diffusion of air and water vapor. If the material permits the diffusion of air, the air will reach a point in the covering where the temperature is below the dew point. The condensed water will gradually accumulate until the covering becomes saturated, which will increase the conductivity and perhaps lower the mechanical strength of the covering until it becomes worthless.

2 ● Compute the total annual heat loss from 165 ft of 2-in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained from Table 8, Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 deg. By interpolation of Table 1 between temperature differences of 157.1 F and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 deg is found to be 1.677 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire = 1.677 × 169.4 × 165 (linear feet) × 4000 (hours) = 188,000,000 Btu.

3 ● Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in Question 2. If the system is operating at an over-all efficiency of 55 per cent determine the monetary value of the annual heat loss from the line.

The cost of heat per 1 million Btu supplied to the system = 1,000,000 × 11.5 (dollars) ÷ 13,000 (Btu) × 2000 (lb) × 0.55 (efficiency) = $0.804. The total cost of heat lost per year = 0.804 × 188 (million Btu) = $151.15.[4]

4 ● If the steam line given in Question 2 is covered with 1-in. thick 85 per cent magnesia, determine the resulting total annual heat loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

By interpolation of Table 9 between temperature differences of 157.1 F and 227.7 F, the coefficient of transmission for 1-in. magnesia on a 2-in. pipe is found to be 0.525 Btu per hour per square foot of pipe surface per degree temperature difference at a temperature difference of 169.4 deg. The total hourly loss per square foot of insulated pipe will then be 0.525 × 169.4 = 89.04 Btu. From Table 5 the area per linear foot of 2-in. pipe is

[4]A closely approximate solution of this problem may be quickly made by use of the estimating chart given in Fig. 1.

found to be 0.622 sq ft. The total annual loss through the insulation = 89.04 × 0.622 × 165 (linear feet) × 4000 (hours) = 36,550,000 Btu. The annual bare pipe loss as determined in the solution of Question 2 was found to be 188,000,000 Btu. The saving due to insulation is then 188,000,000 − 36,550,000 = 151,350,000 Btu per year.

From the solution of Question 3 it was found that the heat supplied to the system cost $0.804 per million Btu; therefore, the monetary value of the saving = 0.804 (dollars) × 151.35 (million Btu) = $121.69, or 81.2 per cent of the cost when using uninsulated pipe.

5 ● The manufacturer's list price for 85 per cent magnesia insulation is $0.36 per linear foot for 1-in. (standard thick) material to cover a 2-in. pipe. Determine the period of time required for the saving found in Question 4 to pay for the cost of the insulation if it can be purchased and applied at 80 per cent of list price (20 per cent discount).

The applied cost of insulation = 165 (linear feet) × 0.36 (dollars) × 0.80 (net) = 47.52. Since the annual saving as found in Question 4 amounts to $121.69, the insulation will pay for its cost in 47.52 ÷ 121.69 = 0.3905 years; in other words, the cost will be repaid 2.56 times by the saving obtained in one heating season.

6 ● The conductivity of magnesia insulation is 0.455 at the mean temperature which will result under the conditions of Question 4. Estimate the most economical thickness of magnesia for application on the pipe when operating under the conditions which are given in the foregoing problems and when a 20 per cent return is required on the investment for insulation.

Use chart given in Fig. 3. Begin at the left bottom margin and proceed successively as shown by the dotted line example to the following essential data which are collected from the problems previously given:

4000 hours operation per year.
$0.804 value of heat, dollars per million Btu.
169.4 deg temperature difference.
0.455 conductivity of insulation.
20 per cent discount from list, cost of insulation.
20 per cent fixed charges, return on investment.
2-in. pipe size.

Solution of the problem by use of Fig. 3 results in a required thickness of approximately 1.05 in. The nearest commercial thickness procurable is standard thick (1⅓₂ in.) magnesia.

(It is of interest to note that the use of Fig. 3 will generally result in solutions which, for all practical purposes, agree closely with the specifications for thicknesses given in Table 18.)

7 ● Determine the minimum thickness of wool felt insulation having a conductivity of 0.30 necessary to prevent condensation of moisture on a 4-in. pipe carrying cold water at a temperature of 40 F when the surrounding air reaches maximum conditions of 90 F with a relative humidity of 90 per cent.

The difference between the temperature of the pipe and the surrounding air is 90 − 40 = 50 deg. For quick estimating purposes use the chart given in Fig. 2. Enter this chart at the lower left margin on the 90 per cent relative humidity line and proceed horizontally to the right to intersect the 90 deg air temperature line. Project a line up to the 50 deg temperature difference line, and then horizontally to the right to the intersection with the 4-in. pipe size line. From this point proceed down to intersect the 0.30 line which denotes the conductivity of the insulation. Directly opposite this point of intersection the correct thickness of insulation is read from the scale on the lower right margin. This chart solution denotes that wool felt 2.4-in. thick is sufficient to prevent condensation. The nearest commercial thickness procurable is 2½ in.

For prevention of condensation as well as for protection against freezing, if the thickness determined theoretically cannot be had, it is better to apply the next greater thickness procurable rather than to use any lesser thickness because an additional factor of safety is thus obtained.

Chapter 40

ELECTRICAL HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Steam Heating, Electric Hot Water Heating, Heating Domestic Water Supply, Industrial Heating, Reversed Cycle Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Power Problems, Insulation

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations, accelerated in many territories by the load building efforts of the utilities which usually include reduced rates to encourage such installations. Electrical heating has a logical place in the heating industry because of its features of flexibility, cleanliness, safety, convenience and ease of control. Electrical heating practice has many basic principles in common with fuel heating, but there are also important differences. When heat units are delivered to each room by wire, no combustion process is necessary, either at a central plant or at the individual room units. The maximum output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the maximum total load on an electrical heating system is the total wattage of connected electric heaters, regardless of weather conditions. The real obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself. Because the heat units produced electrically are more costly, their conservation is of more relative economic importance than with fuel heating, so that sponsors of electric heating give greater attention to temperature-insulated building construction, and to economy by accurate controls.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, because 100 per cent of the energy applied in the resistor is always transformed into heat. In electric heating practice the engineer need not be concerned about efficiencies of heat production, but rather about efficiencies of heat utilization. It is the engineer's problem to distribute the electrically

produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of terms used in fuel heating are given in Chapter 45. The following terms apply particularly to electric heating:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS AND HEATING ELEMENTS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

Commercial electric heating elements are made in many types. Some have resistors exposed to the air being heated. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. This type of element is used extensively for operation at high temperatures when radiant heat is desired, also at low temperatures for convection and fan circulation heating, especially in large installations.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in some types of convection air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. Cloth fabrics woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes such as heating pads and aviators' clothing.

ELECTRIC HEATERS

Electric heaters may be divided into three groups, conduction, radiant and convection.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature incandescent heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. A typical radiant heater is shown in Fig. 1.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors, They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from

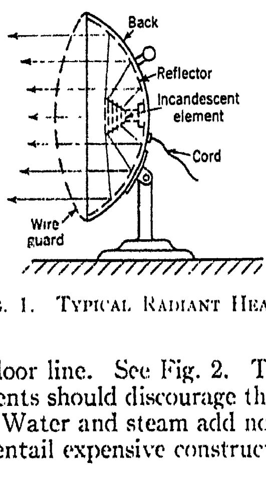


FIG. 1. TYPICAL RADIANT HEATER

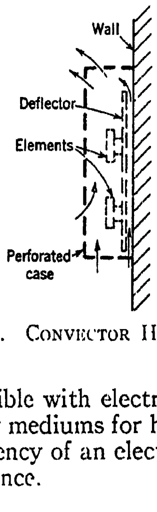


FIG. 2. CONVECTOR HEATER

the floor line. See Fig. 2. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

UNIT HEATERS

Unit electric heaters include a built-in fan unit which circulates room air over the heating elements. Heaters of this type are manufactured in many designs and sizes, and can be located in much the same manner as steam unit heaters.

Electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where the power rate for electric heating is found to be favorable. The best location of the heater depends upon local circumstances as they can be mounted either on the ceiling to direct the air downward, on the side wall about 7 ft from the floor, or at the floor line. Variations in design are necessary for different locations, but typical arrangements are indicated in Figs. 3, 4, and 5.

The arrangement of the wiring circuits is very important for electric unit heaters. In principle they are all the same and include as essential

elements an automatic control panel, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnet coil of the control panel and with the hand switch and thermostat. A typical wiring diagram is shown in Fig. 6. This applies to a single phase power supply, but for 3 phase the only difference is to have a 3 pole panel and a heater arrangement for 3 phase connection.

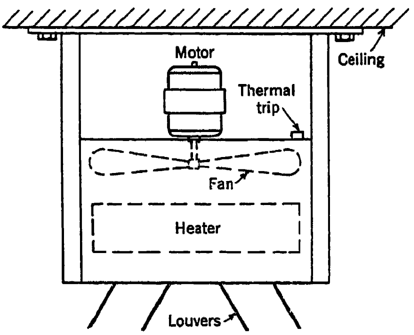


FIG. 3. CEILING MOUNTED UNIT HEATER

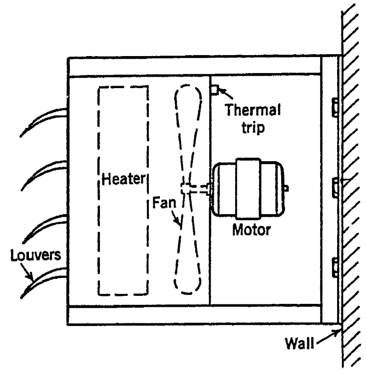


FIG. 4. WALL MOUNTED UNIT HEATER

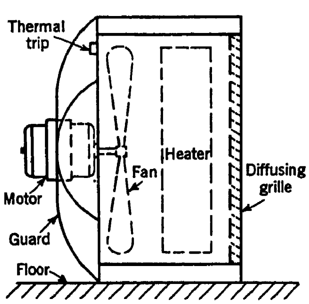


FIG. 5. FLOOR MOUNTED UNIT HEATER

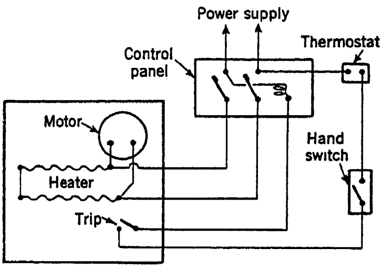


FIG. 6. WIRING DIAGRAM FOR UNIT HEATER

Portable unit heaters are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete air conditioning system. They can be used in the same manner as steam served heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evapori-

zation of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer. See Chapter 21.

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure as a change in air volume flowing over steam coils does not greatly affect the temperatures of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered.

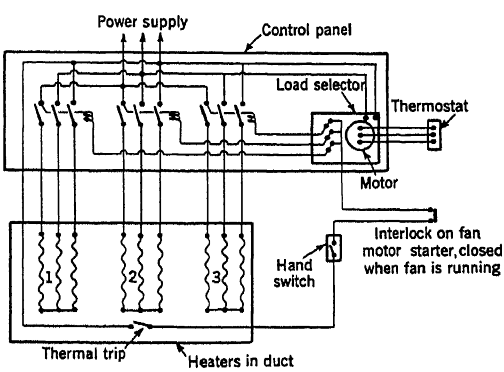


FIG. 7. WIRING DIAGRAM OF A MODULATING CONTROL FOR A FAN SYSTEM

This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic modulation to vary the electrical heat input and synchronize it properly with the air flow has been successfully applied to central fan systems. Electric booster heaters are often useful in balancing a system in which the air has been heated with steam coils.

A typical wiring diagram of an automatic modulating system for central fan heating is indicated in Fig. 7.

ELECTRIC STEAM HEATING

Electric steam heating differs from fuel heating only in the use of *electric boilers* to generate steam. Electric steam boilers are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used on either direct or alternating current since the heat is delivered to the water by

contact with the hot surfaces. To lessen the likelihood that the heating elements will burn out, they should be of substantial construction, with a low heat density per unit of surface area. Provision should be made for cleaning off deposits of scale which restrict the heat flow. A typical

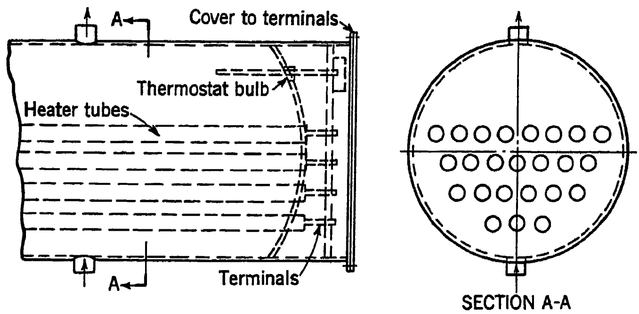


FIG. 8. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

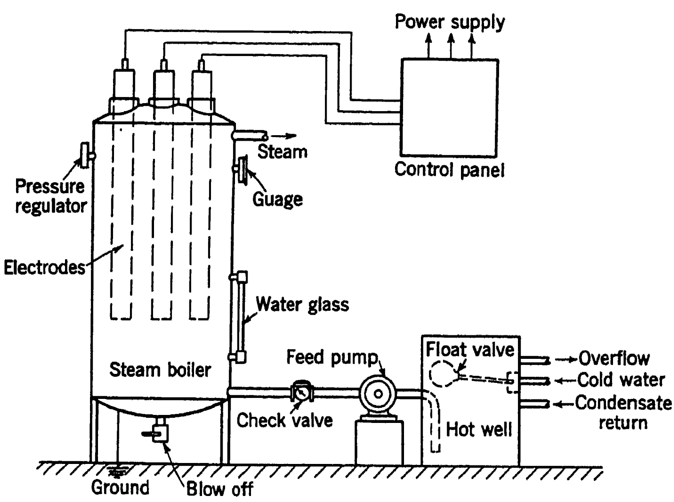


FIG. 9. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

resistance type of hot water or steam boiler is shown in Fig. 8. Large electric boilers are usually of the type employing water as the resistor. Only alternating current can be used, as direct current would cause electrolytic deterioration. Large boilers of this kind have electrodes immersed in the water where heat is generated directly. Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical

to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous. A diagrammatic arrangement of an electrode boiler is shown in Fig. 9.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. The use of insulated water storage tanks in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements connected to line with automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard individual radiator or fan-served indirect type or with provisions for the heating and humidification phases of an air conditioning system. A system of this kind requires very careful design to avoid excessive overall radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical hot water heating boiler is illustrated in Fig. 8.

HEATING DOMESTIC WATER SUPPLY [1]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable, and in many sections of the country low electric rates have been established by the electric utilities to secure this load. In some districts, rate schedules divide the current used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed out, without too frequent demands on the regular heating element which takes the higher domestic lighting service rate. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all service if an electric water heater is installed.

A comprehensive survey covering United States and Canada shows a rapidly growing use of electric water heaters, although the per cent of saturation, based on the total number of domestic power customers, is still low. Public acceptance is effected by the cost of other competitive fuels, by the electric power rate, and by the temperature of the cold water supply.

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[1]Test Results of Electric Water Heaters, by C. G. Hillier (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1936, p. 632).

Fourth Annual Survey, by B. J. Martin (*Electric Light and Power*, March, 1937).

Competition with other fuels, especially gas, seems to be the major controlling factor. The first cost of electric storage heaters is also greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

It is often desirable to connect an electric heater to a residential system having a coil in the fire-box of the furnace. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 10, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to the electric heater serves as a tempering tank, absorbing considerable warmth from the basement air and requiring the use of less energy in the electric heater.

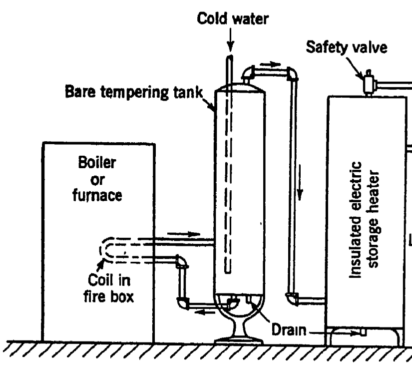

FIG. 10. PIPING ARRANGEMENT FOR CONNECTING ELECTRIC WATER HEATER TO FIRE-BOX COIL

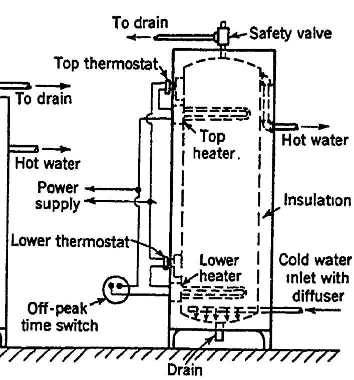

FIG. 11. DOMESTIC HOT WATER HEATER FOR OFF-PEAK SERVICE

A typical domestic hot water heater as shown in Fig. 11 is arranged with upper and lower heating elements for the usual type of off-peak heating service for which most utilities have especially attractive low power rates. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that in case the supply of hot water in the tank becomes exhausted the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

INDUSTRIAL ELECTRICAL HEATING

Electric heating elements have been successfully developed for industrial work such as annealing, brazing, carburizing, enameling, forging, ceramic firing, hardening, metal melting, nitriding and process heating. Industrial ovens and furnaces where precise control of temperature is necessary can be very successfully operated with electric resistance ele-

ments at temperatures as high as 1800 F. For higher temperatures the electric arc or high frequency induction methods are often used. Electric heaters for heating oil to high temperatures for secondary circulation in process work are used as a substitute for superheated steam. Special oil can be electrically heated as high as 700 F and pumped at a pressure just sufficient to cause flow. When used in heating coils or jacketed vessels, this gives a safe and convenient automatic system for moderate-sized installations. Pitch, waxes, and many chemicals are successfully heated by electricity, but require careful design and adequate automatic control.

REVERSED CYCLE REFRIGERATION [2]

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service. For a detailed description of this cycle see Chapter 24.

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electrical heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a building comfortable. Likewise, such electrical heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down

[2]Cooling Homes, A Field for Refrigeration, by A. R. Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926.

The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T. G. N. Haldane (*Electric Review*, Vol. 105, p. 1161-1162, December 27, 1929, and *I. E. E. Journal*, Vol. 68, p. 666-675, June, 1930).

Edison Building Heated and Cooled by Electricity, by H. L. Doolittle (*Power*, Vol. 74, p. 384, September 8, 1931).

House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R. E. Marbury (*Electrical World*, Vol. 100, p. 828, December 17, 1932).

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 307).

Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L. Galson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935, p. 497).

can be conveniently cared for electrically. Fan type unit heaters delivering warm air at the floor zone are very effective.

CONTROL

Because the efficiency of electric heat production is the same for large or small units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Radiant heaters are usually controlled manually but new methods for automatic control are described in Chapter 41 on Radiant Heating. For all convection and fan circulation heaters thermostatic control is essential for economical operation. For duct systems having a variable volume of air flow. the electric heater control must automatically vary the heat input in coordination with the changes in air volume and demand for heat.

CALCULATING CAPACITIES

The methods of calculating heat losses outlined in Chapters 6, 7, and 8 may be used for electric heating exactly as for fuel heating. The total heat requirements in Btu per hour may then be converted into the electrical rating of an equivalent heating system by using the equation:

$$\frac{\text{Total Btu per hour}}{3415} = \text{kw rating of required electric heating} \qquad (1)$$

For comparison with steam radiation:

$$\frac{\text{3415 Btu (one kwhr)}}{240} = \text{14.2 sq ft of steam radiation}$$

While many empirical rules based on cubic contents, floor areas, etc., are used in steam heating practice, they should never be used to determine size of equipment for an electrical heating installation.

POWER PROBLEMS

The first point to determine is the cost of the power which is available for electric heating. Unlike fuels, there is no uniform cost for electric power because of the unequal cost of distribution to large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate power plants at uniform loads; hence, even the time of use may affect the cost of power.

Special low rates are sometimes available during certain prescribed hours of use, but wherever the use of power is unrestricted a demand charge based upon the rated connected load of each heating device must form part of the basic rate structure, so that unlike fuel heating, the cost of operating electric heating systems depends not only upon the actual energy used but also upon the demand charge for the available electrical service.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually

the service lines will not permit more than plug-in devices. The Underwriters permit approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles. Where homes have 230 volt service for cooking and water heating, and rates are favorable, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 220, 440 or 550 volts. All polyphase heaters should be balanced between phases.

INSULATION

The value of building construction which incorporates built-in insulation to reduce the outward heat loss in winter and the inward heat gain in summer has been placed in the spotlight by the increasing adoption of complete air conditioning. With electric heating, adequate insulation is very important and will pay even better returns on the investment than for less expensive fuels.

PROBLEMS IN PRACTICE

1 ● Under what conditions are electric heaters most feasible?

a. When electric power rates are low and other fuels high in cost.

b. Where the total required heat is not great and cost of attention tends to offset the higher actual energy cost of electricity.

c. Where saving of space, elimination of a chimney and lower first cost of equipment are deciding factors.

d. Where intermittent and local auxiliary use avoids the necessity of keeping up steam with large losses in long pipe lines.

e. Where accurate local automatic control reduces the total heat losses enough to compensate for the higher energy rate for electricity over other fuels.

f. For isolated or unattended rooms and small buildings where other heat sources are not readily available.

g. For underground rooms and vaults where the return or disposal of condensation is difficult or where freezing may occur at times.

h. Where corrosive conditions make pipe lines expensive to maintain.

i. Where the use of power for heating can be staggard to avoid periods of other use such as large motors, and thus prevent an increase of the basic demand charges for electrical service.

j. For fall and spring use, and as auxiliary to help out other heating systems at important points during extremely cold weather.

k. Where dust, gases, odor, noise, or access for attendants must be excluded.

2 ● On what basis should electrical heating cost be compared with other fuels?

a. Initial investment with interest and depreciation.

b. Operating cost for energy.

c. Saving in repairs and attendance.

d. Economy due to local accurate temperature regulation.

e. Safety, convenience, and cleanliness.

f. Saving in space.

3 ● At what rate for electric power is electric heating feasible?

a. The answer is complicated because operating cost of electric heaters depends upon at least three factors, namely, demand charge, energy charge, and the sliding scale according to the amount of power used for all purposes.

b. For off-peak use which avoids the demand charge an energy rate of 1 cent per kilowatt-hour is often attractive for heating systems of moderate size. Larger jobs usually require a rate of about ¾ cents per kilowatt-hour.

c. For unrestricted heating service for auxiliary purposes an average rate of 2 cents per kilowatt-hour may be satisfactory for small installations.

d. Wherever other factors make electric heating especially desirable the rates may be even higher than those mentioned above.

e. For domestic hot water supply a rate of 1 cent per kilowatt-hour is usually satisfactory.

4 ● What advantages have electric fan unit systems over plain convection heaters?

a. Better distribution of heat to the floor level.

b. Elimination of condensation on machinery, windows, and other cool surfaces.

c. Wider choice as to location of the heater, and reduced cost of wiring connections.

d. The fan can be operated for air circulation only, during periods when heat is not required.

5 ● In a fan type electric heating system, what important features are required that are not needed for a steam system?

a. A heating coil supplied with steam at constant pressure will remain at approximately constant temperature regardless of the amount of air passing over it, but the condensation rate will change. The temperature of an electric coil supplied with a constant amount of energy will rise if the quantity of air is decreased, and fall if the quantity of air is increased. This happens because the input of electrical energy is constant while the input of steam energy varies with the condensation rate.

b. Because the temperature of an electrical coil will rise upon decreased air flow, the Underwriters require the installation of an approved thermal safety trip switch located in the heating chamber to cut off the electrical heating circuit automatically in case the air flow should be interrupted. This switch should remain off until manually reset.

c. Electrical heating elements vary greatly in their capacity for storing heat units after the power is shut off. In a fan system it is very important to use elements having the lowest possible heat storage capacity to avoid overheating motors and improperly operating the thermal safety trip switches when air flow ceases due to normal shut-downs.

d. Because the input of an electric heater is constant for a given rating, it is necessary to provide a modulating electrical control which will automatically compensate for variations in the volume of air flow. This cannot be done by mixing dampers alone as in a steam system.

6 ● What problems must be considered in connection with electric water heating?

a. For the best available power rate consult the Electric Power Company supplying service.

b. The maximum daily and hourly demand for hot water.

c. The size of storage tank necessary to carry over the peak load periods when no reheating is done.

d. The kilowatt rating required to reheat enough water during off-peak periods.

e. Standby radiation losses of the storage tank and hot water piping.

f. Cost of providing electrical supply lines of ample capacity with fuses, meters and switches.

g. Tank materials and design to avoid expensive replacements due to corrosion.

Chapter 41

RADIANT HEATING

Physical and Physiological Factors, Control of Heat Losses, Rate of Heat Production, British Equivalent Temperature, Application Methods, Calculation Principles, Mean Radiant Temperature, Measurement of Radiant Heating

HEATING for health and comfort is generally understood to mean that heat must be supplied in order to control the rate of heat loss from the human body so that the physiological reactions are conducive to a feeling of comfort. In convection heating, it is generally the function of the heating medium to transfer the heat to the air and thence to the occupant of the room, while the primary object of radiant heating is to warm the surrounding surfaces without appreciably heating the air. The difference between convection heating and radiant heating is therefore partly physical and partly physiological. Reference to low temperature radiation is actually not heating at all, except in a secondary sense. Low temperature radiation is produced not to heat the individual in the room, but to reduce the net rate at which the body surface loses heat by radiation.

Comfort requires that heat be removed from the body at the same rate as it is generated by the oxidation of food stuffs in the body tissue. Furthermore, the heat should be dissipated in a manner conducive to the physiological requirements of the human body. Actually the feeling of heat and cold in an individual is not so much a measure of the rate at which body heat losses take place as compared with the heat generated within the body, as it is an indication that the sensation of the body is more susceptible to the manner in which the heat is abstracted from the body. This principle is the basis upon which radiant heating is founded.

CONTROL OF HEAT LOSSES

Heat is transferred from any warm dry body to cooler surroundings principally by convection and by radiation, the approximate total being the sum of the two. Where the body surface is moist as with the human body, there is additional loss of heat through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the difference between the temperature of the body and of the surrounding air, and on the rate of air motion over the body.

The loss by radiation of a given surface depends entirely upon the difference between the temperature of the body and the mean surface temperature of the surrounding walls and objects. This latter temperature is called the *mean radiant temperature* (MRT).

Because these two types of heat losses act in a supplementary manner toward each other, a required rate of heat loss can be secured by having a relatively low air temperature and a relatively high MRT, or *vice versa*. Thus, if the air is reduced from a given temperature to a lower temperature, the amount of heat lost from the body by convection is increased, and this increase can be compensated for by raising the MRT. Similarly, with a higher air temperature the same total heat loss will be maintained by a correspondingly lower MRT.

Within limits the sensation of feeling cold can be avoided in two ways; first, by raising the air temperature surrounding the body, and secondly, by allowing the thermal radiation from warm objects to impinge on the body with sufficient intensity to make up for a lower air temperature.

It is the object of a heating installation to avoid the necessity for human body adaptation and also to provide comfort for those individuals doing the least physical work. While some conditions may take care of the heat loss from the body without controlling the generation of heat within, other conditions stimulate the production of heat within us, which enables the body to respond to the environment and generate more heat to meet the conditions.

Rate of Heat Production

The normal rate of heat production in a sedentary individual is about 400 Btu[1] per hour, or (since the entire surface area of an average adult is 19.5 sq ft) about 20.5 Btu per square foot per hour. When considering radiant heating, it is necessary to calculate the radiation and the convection loss separately. The human body is of complicated shape, and radiation only takes place freely from the exposed outer surface. There are considerable portions of the body which radiate most of their heat to other portions, such as: the legs, arms, lower part of head etc. It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total surface for convection may be assumed as an approximate value of 19.5 sq ft and 15.5 sq ft for radiation.

The heat generated in the average human body is approximately 400 Btu per hour of which 75 per cent or 300 Btu per hour is the approximate value of the heat given off by radiation and convection. While it is difficult to differentiate the exact proportion of these two values, it is found that if the body gives off about 190 Btu per hour by radiation or 12.26 Btu per hour per square foot of radiant body surface, conditions of greatest comfort will result. This leaves 110 Btu per hour to be released by convection, or 5.64 Btu per hour per square foot of convected body surface.

[1]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245).

The loss by evaporation, which depends on the air temperature, air movement and humidity, together with the loss by respiration makes up the balance of 100 Btu per hour. All of these values are relative because the total will vary materially with change of position, occupation, age, race, etc.

The mean normal surface temperature of the human body, taken over the whole area, including not only the exposed skin surface but also surfaces of the clothes and the hair, has been very extensively used as 75 F, particularly in England where radiant heating has been practiced for nearly 30 years. However, results obtained by Aldrich[2] in rooms in which the air and wall surface temperatures were approximately 72 F gave mean values nearer 83 F than 75 F. In both England and America mean wall and air temperatures of 72 F seem to be unwarranted; so it is not unreasonable to assume that a body surface temperature lower than 83 F may eventually be accepted. Some values have already been suggested as being the most suitable for the American climate, but the accepted standard for United States practice must be ultimately derived from research and practical experience.

The mean surface temperature of an inert body which will maintain the optimum heat loss by radiation and convection in a uniform environment of a given temperature may be calculated from fundamental equations for radiation and natural convection by substituting comparable cylinders for the body. While it may be possible to produce effects on a cylinder or any other body of a particular size and shape to estimate similar effects on the human body, it should be remembered that the heat loss from the body varies greatly with movement. Every movement of the body not only alters its shape but also the velocity of the air passing over it and the surface exposed to radiation. This fact makes it difficult to compare the effect of any environment on a cylinder to that of a human body. Heilman[3] gives the following equations:

$$H_r = 0.1723\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$H_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

H_r = heat loss by radiation, Btu per square foot per hour.

H_c = heat loss by convection, Btu per square foot per hour.

T_s = absolute temperature of the body surface, degrees Fahrenheit.

T_w = absolute temperature of the walls, degrees Fahrenheit.

T_a = absolute temperature of the air, degrees Fahrenheit.

$T_m = \dfrac{T_s + T_a}{2}$

D = diameter of cylinder, inches.

e = the ratio of actual emission to black body emission.

If it is assumed that a normal adult has an average height of 5 ft 8 in.

[2] A Study of Body Radiation, by L. B. Aldrich (Smithsonian Miscellaneous Collections, Vol. 81, No. 6, December 1928).

[3] Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transactions, Fuels and Steam Power Section*, Vol. 51, No. 22, September-December, 1929).

and an average body surface of 19.5 sq ft and 15.5 sq ft for convection and radiation respectively, an equivalent effect can be considered on two cylinders 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter respectively.

BRITISH EQUIVALENT TEMPERATURE

The British Equivalent Temperature (BET) is the temperature of an environment which is effective in controlling the rate of sensible heat loss from a sizable black body in still air when the body has a maintained surface temperature equal to that of the human body. The BET is, therefore, a function of both the air temperature and the mean radiant temperature. Its numerical value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and the air. In a non-uniform environment (walls and air at different temperatures) the BET for America is at present considered to be equivalent to that of a uniform environment in which an 83 F surface loses sensible heat at the same rate as it does in the non-uniform environment. As originally defined, the BET was based on a body surface temperature of 75 F, but 83 F has been accepted as giving results more nearly conforming with American practice[4]. Temperatures selected depend on the clothes worn by the individual, which explains why ladies in evening dress desire a higher body surface temperature than a man dressed in evening suit leaving only hands and head uncovered.

For accurate calculations it would be more logical to assume a body surface temperature applicable to the room being occupied, but for general purposes it is considered sufficient to take an average of 83 F for all rooms. The higher the BET the less the heat loss from the body, as the rate of loss in still air is approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no sensible heat loss from a surface at that temperature; so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated.

APPLICATION METHODS

There are several methods of applying radiant heating, as follows:

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe spaced about 6 in. apart (Fig. 1). This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should not exceed about 130 F due to the possibility of cracking the plaster, the area of the panel must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces comfortable and economical results, but offers some slight obstacles when alterations or additions to the building are desirable. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All of the pipes are welded together and tested after erection to a hydraulic pressure of 500 lb per square inch.

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[4]Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

2. *By placing hot water or steam pipes under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. This method is used extensively for schools and hospitals where large quantities of outside air are desirable (Fig. 2). In some cases special floors are con-

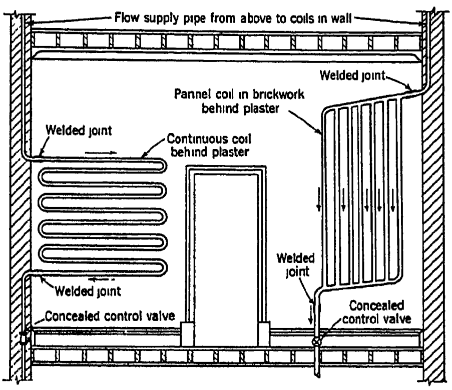

FIG. 1. PIPE COILS LOCATED IN INTERIOR WALL SURFACES

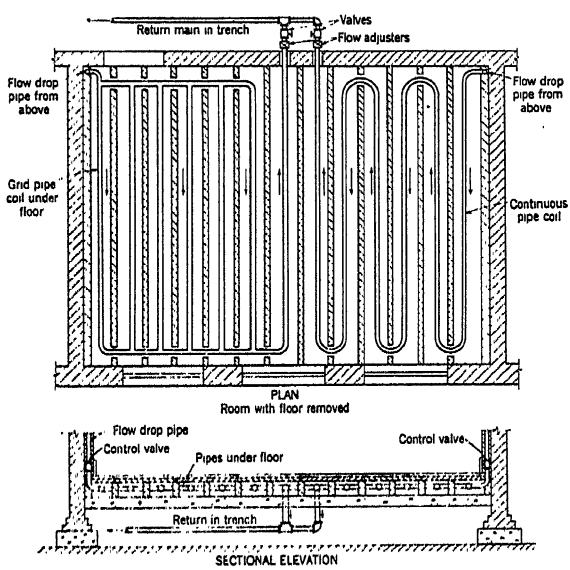

FIG. 2. ARRANGEMENT OF CONTINUOUS PIPE COIL IN FLOOR CONSTRUCTION

structed in sections so that a whole floor can be lifted to examine the pipes. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat. Pipes under the floor may be larger than those embedded in the plaster walls and ceilings.

3. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in method 2. This method while being more

expensive in construction, is effective and quite suitable for cathedrals and large public buildings (Fig. 3). To provide a uniform floor temperature, special consideration should be given to the design of the air ducts so that equal distribution is obtained.

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other parts of the room, or they can be cast into panels to imitate oak or other wood designs. With flat plate panels it is a common practice to use a frame of plaster, wood, metal or composition around the panels to allow for expansion. These plates may be heated with either hot water or steam and connected similarly to an ordinary radiator system.

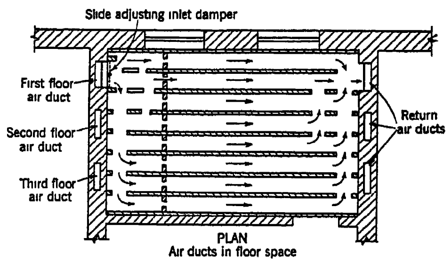

FIG. 3. DIAGRAM OF AIR DUCTS FOR FLOOR HEATING

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the face as desirable. They should not have a surface temperature above 160 to 200 F. Some electric panels have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall.* For this purpose the screen is woven with an electric continuous conductor being the warp and wool or silk being the weft. Such screens are useful to plug in at any position for emergency local heating without taking care of a large room or office.

Note. If all of the heating panel is installed at one end of a large room there may be a marked difference between the BET on the two sides of the body. It is usually desirable therefore that the heat be distributed at different parts in the room so that no uncomfortable effects will be felt from unequal heating.

CALCULATION PRINCIPLES

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine the rate of heat loss from the room by conduction, convection, and radiation when maintained in the desired condition; radiant heating involves the regulation of the rate of heat loss per square foot from the human body.

The first step in the calculations for radiant heating is to ascertain the necessary mean radiant temperature (MRT): next, the size, temperature, and disposition of the heating surfaces required in the room to produce this MRT are estimated; and after this the determination of the convective heat is made.

Mean Radiant Temperature

If the whole of the interior surface of a room were at the same temperature, this temperature would represent the MRT. Such a condition

TABLE 1. TOTAL BLACK BODY RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126.6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula

$$Q = e\left(\frac{0.1723 \times T^4}{100{,}000{,}000}\right)$$

where

Q = total black body radiation, Btu per square foot per hour.
e = emissivity.
T = absolute temperature, degrees Fahrenheit.

seldom exists, however, since the actual surface temperature in any heated space having surfaces exposed to the outer air varies greatly for different sides of the enclosure. It is therefore necessary to ascertain by calculation the mean of these interior surface temperatures.

The mean temperature in this sense is not the arithmetic average of the actual thermometric temperatures of the surfaces, but the temperature corresponding to the average rate of heat emission per square foot of surface. The temperature corresponding to this mean emission can be taken from Table 1. Conversely, the emission at different temperatures and also the emissivity factors can be obtained from this table. For instance, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero if the emissivity of the surface is 0.9.

If the area in square feet of each part of the space is multiplied by the emission value corresponding to its actual temperature, and these products are added together, the gross amount of radiant heat discharged into the room by the wall surface per hour is obtained. This quantity, divided by the total interior surface, gives the average amount of heat coming into the room from the surface of the walls per square foot of surface per hour.

Interpolating in Table 1, the total radiation from a surface at 83 F for an emissivity of 0.95 is 142 Btu per square foot per hour. The difference between 142 Btu and the average amount of heat coming into the room is the amount which will be lost per square foot per hour by radiation from a body at 83 F. If a rate at which it is desired that heat be lost from the body by radiation and convection be assumed, the mean radiant emission from the walls required to give the desired result can be determined from Table 1, as can also the required air temperature for the corresponding convective effect.

The determination of the amount of radiant heating surface needed in a room requires knowledge of the climate, the type of structure, the type of heating, and the surface temperature of the walls. This problem can be solved only on an empirical basis. After some experience, however, it is possible to estimate these variables with a considerable degree of accuracy for any climate or construction.

Assume that a mean radiant temperature of 65 F is desired. Table 1 shows that with all the walls at this temperature, and with an emissivity of 0.95, the gross heat emission is 124 Btu per square foot per hour. The total emission of radiation into the room from that surface would therefore be $A \times 124$, where A is the total inside area of the room. This is the *desired* emission.

If the whole area be divided into a number of different parts which are each at a uniform temperature—a_1, a_2, a_3,—and each is multiplied by the value of the heat emission corresponding to that temperature, and if all these products are added together, their sum will represent the total *actual* emission of radiation into the room at these temperatures without the aid of any hot surface.

The difference between the desired emission and the actual emission represents the additional heat which must be supplied by the hot surface. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. This emission is divided into the additional amount of heat needed, adjusted for the fact that the heating units will shield the walls behind them, and the quotient obtained will be the area of the required heating surface.

It is evident that this method of calculation is approximate, and depends for its accuracy on a correct estimate of the ultimate surface temperatures attained by the actual wall surfaces.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection, and thereby to determine whether this amount of convected heat will warm entering ventilating air to the temperature maintained. If it will not, additional convection surfaces must be introduced to make up the deficiency.

TABLE 2. SURFACE AREAS, TEMPERATURES AND EMISSIONS FOR A ROOM OF 5760 CU FT

	AREA SQ FT	ASSUMED SURFACE TEMPERATURE (DEG FAHR)	HEAT EMISSION (BTU PER SQ FT PER HOUR)	TOTAL HEAT EMISSION FROM AREA (BTU PER HOUR)
External Wall	297	50	110.6	32,850
Glass	279	45	106.5	29,710
Inner Wall	480	55	115.1	55,250
Ceiling	480	55	115.1	55,250
Floor	480	55	115.1	55,250
Total	2016			228,310

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly representative of American practice with well-built walls, and are based on an emissivity of 0.95 which approximates that of most paints and building materials.

The mean radiant temperature of the room is 228,310/2016 = 113.2 Btu per square foot per hour which, as seen from Table 1, corresponds to an MRT of 53 F for an average emissivity of 0.95.

For an average individual having a body surface area of 15.5 sq ft under conditions of comfort with a body surface temperature of 83 F, the heat given off by radiation when calculated by means of Equation 1 is 217 Btu per hour, or 14 Btu per square foot per hour. This corresponds to an environmental emission of 142 − 14 = 128 Btu per square foot per hour, and, according to Table 1, to an MRT of 69.2 F.

If this body be placed in the room described, it will lose heat at the rate of 15.5 × (142 − 113.2) = 446 Btu per hour. This loss is 229 Btu per hour more than the 217 Btu per hour calculated, or 14.77 Btu per square foot per hour, more than the rate of heat loss for comfort.

In order to determine the amount of radiating surface necessary to maintain the MRT at 69.2 F, assume the surface temperature of the hot plates to be installed to be 160 F, which is approximately the temperature they would have if heated by hot water.

The 2016 sq ft total area of the surfaces of the room multiplied by 128 which is the emission in Btu per square foot per hour necessary to maintain a body surface temperature of 83 F, gives a total desired emission of 258,048 Btu per hour. It is necessary to supply enough radiant heating surface to increase the total actual mean radiant heat emission by the room from 228,310, as shown in Table 2, to the 258,048 Btu desired. The additional heat needed is the difference between these figures, or 29,738 Btu. Since, from Table 1, the emission per square foot at 160 F is 238.8 Btu, the required radiant heating surface needed is 29,738/238.8 = 124 sq ft. The effect of this surface suitably placed would be to raise immediately the mean radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

The calculation may be simplified by preparing tables showing, at the usual temperatures, the area of hot surface required to bring each square foot of actual wall surface at various temperatures up to a general standard of from 60 F to 70 F. It would then be necessary only to multiply the respective areas by the appropriate factors, and to add the results, to obtain the required total.

MEASUREMENT OF RADIANT HEATING

Convection heating, having as its object the raising of the air temperature to a specified degree, must be measured by thermometric methods which indicate essentially the air temperature, and not the rate of heat loss from the human body. Radiant heating, having as its object the control of the rate of heat loss from the human body, can be measured

only by methods which basically are calorimetric, that is, which measure directly the rate of heat loss from an object maintained at the temperature of the body, irrespective of air temperature.

The apparatus for this purpose consists essentially of a hollow sphere, or cylinder, containing a fluid which can be maintained accurately at the accepted mean surface temperature of the human body, with an accurate means of measuring the rate of heat supply required to maintain the temperature at that exact point. The latter measurement can be made with sufficient accuracy by electrical methods. Although a definite BET is desirable, the mean radiant and air temperatures may both vary, provided the heat loss by radiation and convection from a surface at 83 F is maintained at the correct proportion or within reasonable limits.

This instrument, the *eupatheoscope*, can readily be adapted as a thermostat by electrical control to shut off or turn on heat when the critical temperature of 83 F or any other predetermined temperature on the surface of the vessel is increased or decreased. A modification of the instrument is called the *eupatheostat.*

Another instrument for maintaining comfort conditions is at present available only in a model adapted to British practice as it is designed for a temperature of 75 F. It consists of a blackened copper sphere of approximately 6 in. diameter in which is housed a cylindrical sump containing a volatile liquid. In operation, a small electric heating coil drawing about 5 watts creates in the sphere a vapor pressure which is constant as long as the heat losses from the sphere are standard. If the temperature of the air or the MRT becomes too high for comfort, a greater pressure is created, owing to a smaller loss of heat from the sphere. This increase of pressure acts on a diaphragm and shuts off the supply of heat to the room.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature.*

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PROBLEMS IN PRACTICE

1 ● Where did radiant heating derive its name?

The term radiant heaters was introduced about 28 years ago to designate flat heating surfaces made to give off practically all their heat by radiant ether waves instead of relying on convected warm air.

2 ● What is actually meant by radiant heating and what are its underlying principles?

The term radiant heating now applies to methods of heating where, instead of heating the air in a room to a predetermined temperature, flat heating surfaces are placed in a room so that the average effective temperature of walls, ceiling, glass and floor surfaces exposed to the body is just sufficient to prevent the body losing too much heat by radiation. It takes into consideration that the body generates more heat than it requires, so that it does not require any heat from without. The surplus heat, however, must be given off according to the physiological requirement of the body.

3 ● What kind of heating surfaces are in general use?

The heating units may have flat iron surfaces heated with steam or hot water and placed in side walls or under windows, or they may be supported on the ceiling and suitably decorated and connected as ordinary steam or hot water radiators. Hot water pipes may be embedded in the floor, walls or ceiling, and when in the floors they may be covered with concrete and wood blocks or other suitable material; the finish of the surface being more important than the composition of the material. When in the ceiling or walls, they can be covered with plaster to harmonize with the rest of the room. Electrical radiant heaters are made by embedding resistance elements in porcelain, or electric conductors may be woven into thick paper and fastened to the walls and ceilings, electric wires may be woven with tapestry to form portable screens for local heating.

4 ● What surface temperatures are generally used?

Where hot water pipes are embedded in plaster, the surface temperature varies from 90 to 130 F. Where flat iron plates are used these may vary from 140 to 220 F. With electric resistances embedded in porcelain the surface temperature may vary from 200 to 500 F. High surface temperatures are not recommended.

5 ● What kind of heat rays are commonly generated for radiant heating?

All heat rays are generally assumed to be the same as light rays; they travel at the speed of light, but they are invisible and longer. The rays used in heating are 0.00005 to 0.0001 in. long, compared with invisible red rays of about 0.000027 in.

6 ● When and why does the human body feel cold?

The body feels cold not only when it loses heat at a greater rate than it can generate it but also when heat is abstracted from the body disproportionately. Since the human body generates more heat than is necessary, it is only necessary to provide conditions that will regulate the correct ratio of losses; the provision of suitable radiant heating surfaces is one way to establish these conditions.

7 ● Why is the heat loss from the body by radiation important?

The heat loss by radiation is proportional to the fourth power of the temperature difference between the surface of the body and the average surface temperature of the surrounding walls, windows, etc.; whereas, for convection losses, it is only proportional to the 1.25 power.

8 ● What is the approximate relation for heat losses?

Heat losses from the body when in a sedentary position are approximately as follows: radiation 49 per cent, convection 23 per cent, evaporation 15 per cent, respiration 11 per cent, and miscellaneous 2 per cent. Actually, it depends upon age, environment and other conditions.

9 ● What generally is the air temperature necessary to give equal comfort effect for sedentary conditions?

With radiant heating, 64 to 66 F. With convection heating, 70 to 72 F.

10 ● Why is there a saving in fuel consumption with radiant heating?

A saving is effected because the differential between inside and outside temperature is much less for radiant heating. Less ventilating air is necessary and this can be supplied at a much lower temperature.

11 ● Describe how to calculate the required amount of radiant heating surface.

a. Obtain the mean heat emission in Btu per square foot per hour for room surfaces X, using values in Table 1, and surface temperatures as shown in second column of Table 2.

b. Deduct X from 142 (142 being the emission per square foot given off by the human body at 83 F surface temperature) = Y in Btu per square foot per hour.

c. From (142-X) deduct 11.1 (11.1 being the average radiation which the human body should lose per square foot for comfort conditions) = (142-X-11.1) = Z.

d. Multiply total interior surface of room by Z and divide by the emission per square foot from radiant heater, giving the surface S of radiant heater in square feet.

12 ● Give a simple formula to calculate radiant heating surface required, and explain.

$$S = \frac{(142 - X - 11.1)\,A}{B}$$

where

S = surface of radiant heater, square feet.
142 = Heat emission, Btu per square foot per hour which the human body would give off at 83 F, with surroundings at absolute zero.
X = mean heat emission, Btu per square foot per hour from surfaces of room.
11.1 = heat emission, Btu per square foot per hour from human body.
A = total surface, square feet of walls, ceilings, windows, etc., in room.
B = heat emission per square foot from radiant heater surface.

13 ● What natural evidence have we that air temperature alone is no criterion of comfort and that radiant heat affects the body more quickly?

When standing in the sunshine on a cool spring day, a person feels perfectly comfortable, but when a cloud passes over the sun, he instantly feels much cooler as the shadow reaches him. A shielded thermometer recording the temperature of the air shows no reduction in air temperature in so short a period, so that the person actually feels a sensation of cold which an ordinary thermometer cannot register. This shows that light and heat rays are shut off simultaneously and travel at the same speed; it also proves that radiant rays affect the comfort of the body quicker than air temperature does.

Chapter 42

DISTRICT HEATING

Piping Distribution, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Conduits for Piping, Pipe Tunnels, Building Service Connections, Steam Consumption, Fluid Meters and Metering, Rates

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories and for the design of heating systems for buildings which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

PIPING DISTRIBUTION

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Any unusual requirements such as those for process steam should be individually calculated.

The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating

peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend, in part, upon whether or not it has been passed through electrical generating units. If it has, the pressure will be considerably lower than if live steam, direct from the boilers, is used. The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, and (3) simpler problems in pressure reduction at the buildings. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators. The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\ W^2L\left(1 + \frac{3.6}{D}\right)}{dD^5} \tag{1}$$

where

P = pressure drop, pounds per square inch.
W = weight of steam flowing, pounds per minute.
L = length of pipe, feet.
D = inside diameter of pipe, inches.
d = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 16.

Information on provision for expansion will be found in Chapter 18.

In general, return lines when installed follow the contour of the land, and Table 1 gives sizes of return pipes for various grades. It is evident that at points where the grade is great, smaller pipes can be installed.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without effecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural steel set in concrete.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft						
	6″	1′	2′	3′	5′	10′	20′
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	....
8	190000	277000	404000	508000	660000	938000	....
10	344000	498000	724000	900000	1190000	....	....
12	555000	798000	1148000	1499000	1990000	....	....

[a]Size of pipe should be increased if it carries any steam.

In laying out conduits of this type the following points should be borne in mind:

1. An expansion joint, offset, or bend should be placed between each two anchors.

2. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart.

3. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.

4. For longer lines, manholes must be located according to judgment and depending upon the expansion value of the type of expansion joint or bend that is used. The minimum number of manholes will be required when an expansion bend or an anchor with double expansion joint is placed in each manhole and the pipes are anchored midway between manholes.

5. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic pressure should be one-and-one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around

the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at *A*. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At *B* and *D* in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation.

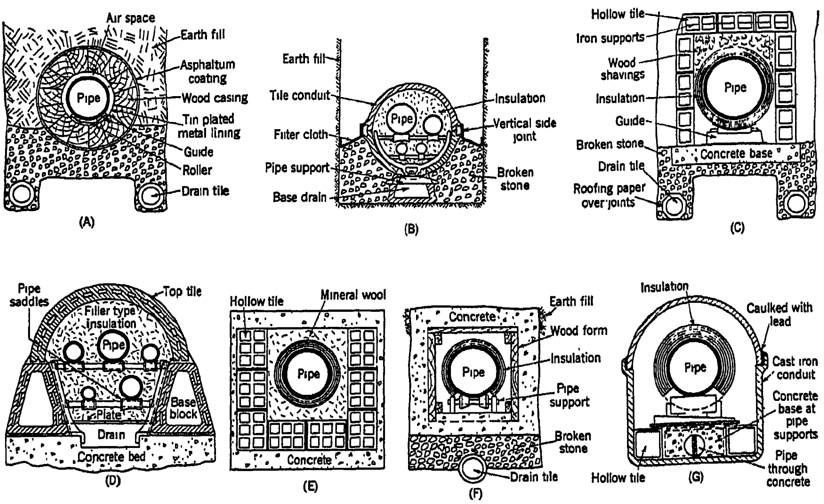


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in

separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a waterproof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required

to accommodate miscellaneous other services or provide underground passage between buildings.

BUILDING SERVICE CONNECTIONS

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer.

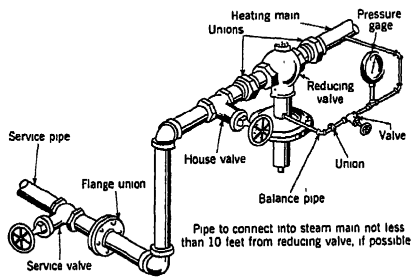


FIG. 2. CONNECTIONS FOR REDUCING VALVES OF SIZE LESS THAN 4 IN.

There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

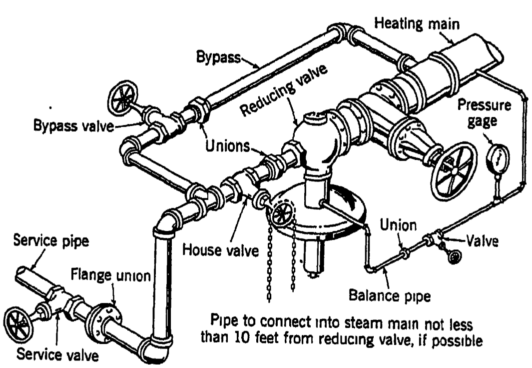


FIG. 3. CONNECTIONS FOR REDUCING VALVES OF SIZE 4 IN. AND LARGER, AND FOR EXPANDED VALVES

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. This latter construction permits the

operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve, usually furnished by the utility company,

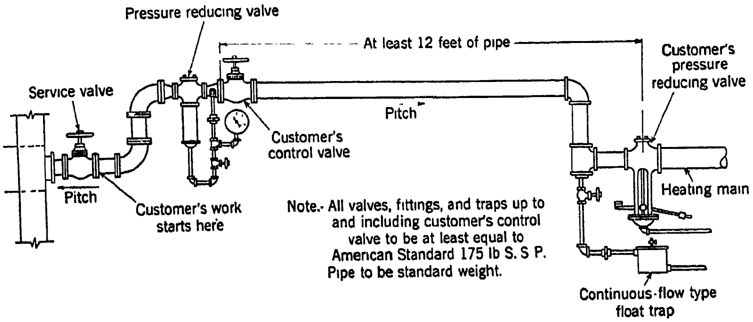


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING CONDENSATION METER

effects the initial pressure reduction. The second reducing valve, usually furnished by the customer, reduces the steam pressure to that required.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in

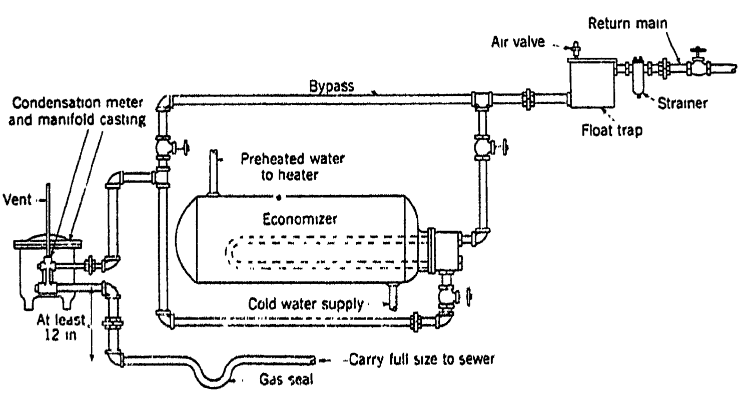


FIG. 5. RETURN PIPING FOR CONDENSATION METER

some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very

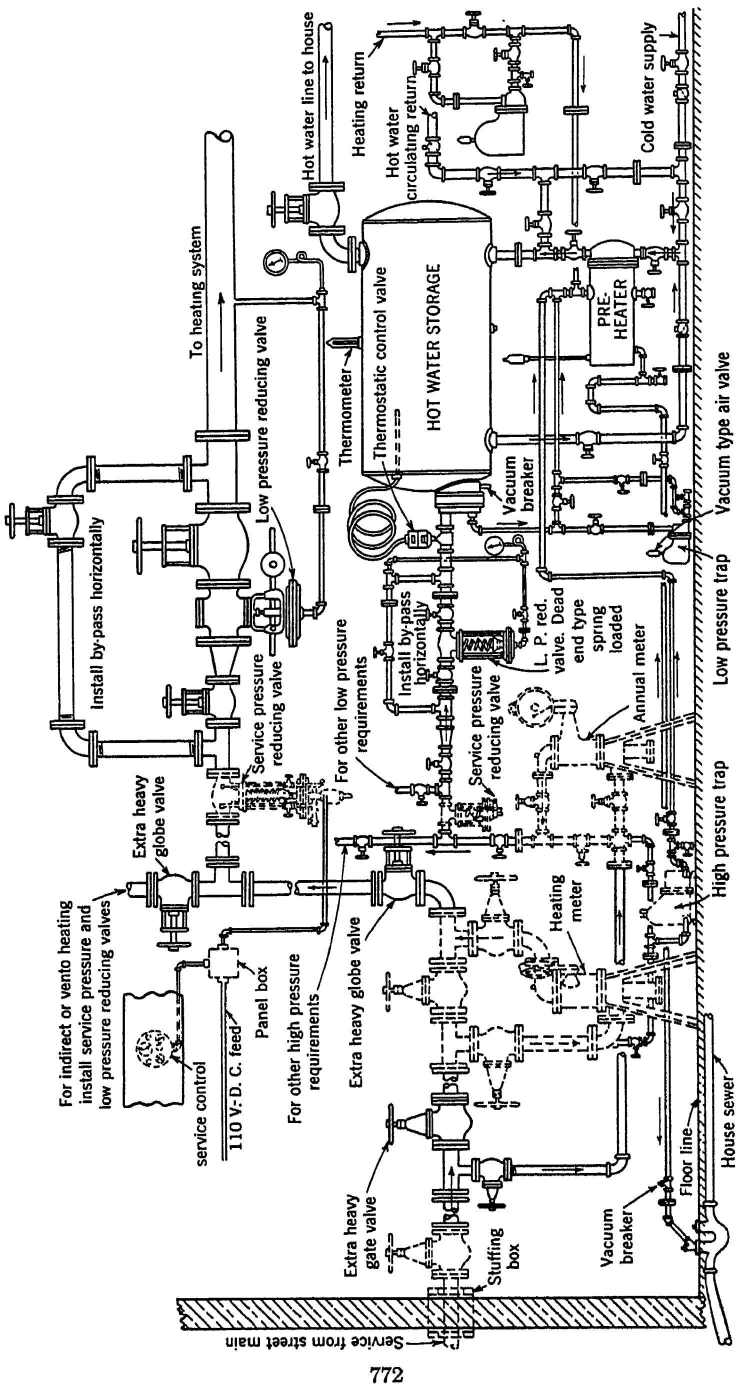


FIG. 6. TYPICAL SERVICE INSTALLATION

cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling radiator, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the preheater on its way to the meter. The supply to the hot water heater passes through the preheater, absorbing heat from the condensation. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the preheater and the water heater proper, not at the preheater inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the preheater should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensation if storage capacity is provided for the preheated water. Frequently a type of preheater is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of thermostats of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating thermostat. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiators either throughout the 24 hours of the day or during the day-

ime hours only. The setting of a switch may provide no service, con-inuous service, or periodic service. For the latter, by means of several ntermittent settings, steam will be supplied during each period in in-rements of a certain number of minutes for each successive setting of the witch, steam being shut off during the balance of the period. These ettings afford from 15 to 80 per cent of the maximum heating effect equired on days of zero temperature. A night switch with a variety of ettings may be adjusted so as to maintain throughout the night the ntermittent supply called for by the day switch setting, or may be set to nterrupt the operation of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected y the operating engineer.

FLUID METERS

No one thing has contributed more to the advancement of district eating than the perfection of fluid meters, which may be classified as ollows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the stream and isolated by alternately illing and emptying containers of known capacity.

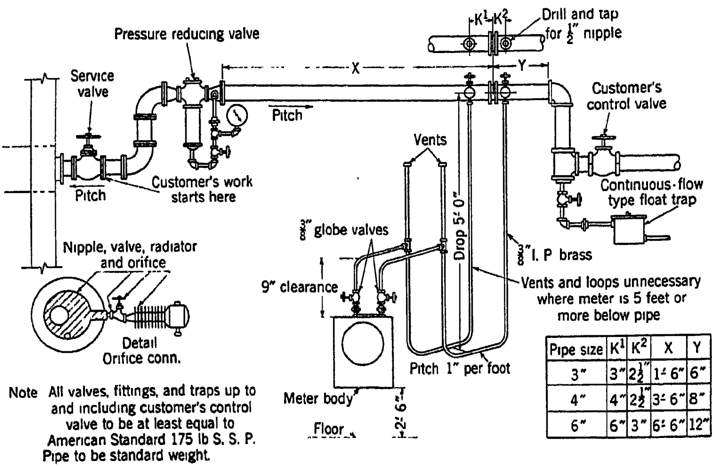
2. *Differential Meters:* The fluid does not pass in isolated separately-counted quan-ities but in a continuous stream which may flow through the line without actuating he primary device of the meter. In the differential meter, the quantity of flow is not letermined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional subdivisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.



Pipe size	K^1	K^2	X	Y
3"	3"	2½"	1'-6"	6"
4"	4"	2½"	3'-6"	8"
6"	6"	3"	6'-6"	12"

FIG. 7. ORIFICE METER STEAM SUPPLY CONNECTION

6. Cleanliness of the fluid to be measured.
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow: turbulent, pulsating, or steady.
10. Cost.
    (*a*) Purchase price.
    (*b*) Installation cost.
    (*c*) Calibration cost.
    (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer.
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing.
14. Stocking of repair parts.
15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are of the condensation or flow types.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Steam flow meters are available in many types and combinations, as indicated in the sub-division covering fluid meters on page 774.

The *orifice and plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a

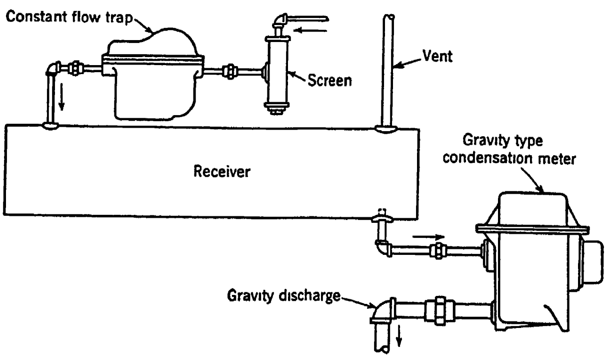


FIG. 8. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 7 shows a typical orifice type meter connection and indicates typical requirements in the installation of this type of meter. Fig. 8 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 9 shows a vacuum installation without a master trap.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.

3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.
4. Meter piping should be kept free from leaks.
5. Sludge should not be permitted to collect in the meter body.
6. The meter body and meter piping should be kept above freezing temperatures.
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM CONSUMPTION

The following factors are used in New York City for the different classes of buildings listed. The factors are based on maintaining an inside tem-

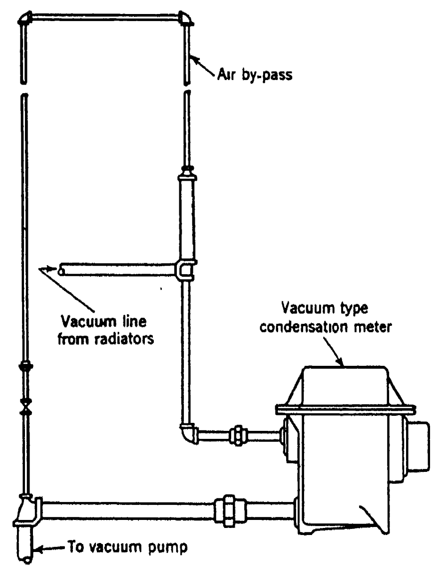


FIG. 9. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

perature of 70 F for certain hours, with a minimum outside temperature of 0 F and an average of 43 F for the heating season of eight months (October 1 to June 1). In this group are six types of buildings:

Manufacturing or commercial loft type where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 a.m. to 5 p.m. No Sunday or holiday use and no night use. *Factor:* 325 lb per square foot of heating surface per season.

Office buildings using steam during daylight hours to maintain 70 F from 9 a.m. to 6 p.m. for approximately 240 days (heating season). No night use. *Factor:* 400 lb per square foot of heating surface per season.

Office buildings using steam during day hours and at night when required to 7, 8 and 9 p.m. (customary where there are stock brokers or banking offices), 240 days. *Factor:* 500 lb per square foot of heating surface per season.

Residences of the block type (not detached) where high-class heating service is required; somewhat similar to apartment buildings. *Factor:* 550 lb per square foot of heating surface per season.

Apartment houses where high-class heating service is required. (Steam off at midnight.) *Factor:* 650 lb per square foot of heating surface per season.

Hotels (commercial type) where very high-class service is required for 24 hours. *Factor:* 800 lb per square foot of heating surface per season.

By assuming one square foot of equivalent heating surface for each 100 cu ft of space heated, which seems a fair ratio in New York City, it is possible roughly to estimate the steam required per cubic foot of space, information which is often more easily obtained than the square feet of heating surface. Additional data on the heating requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association*.

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. The profit need not be divided proportionately among the rate groups, but should be established from a competitive standpoint. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average load to the maximum load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

A. Flat Rates.
  1. Radiator surface charge. *Obsolescent.*

B. Meter Rates.
  1. Straight-line.
  2. Step. *Obsolescent.*
  3. Block.
     (*a*) Class rates.

C. Demand Rates.
  1. Flat demand.
  2. Wright.
  3. Hopkinson.
  4. Doherty (or Three charge).

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate The pounds of steam consumed by a customer are divided into blocks of *M* lb each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefitted at the expense of the others.

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per *M* lb of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a flow meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(*a*) A charge based upon the demand, either estimated or measured;

(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand. They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per *M* lb of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

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PROBLEMS IN PRACTICE

1 ● What is the common method of determining the size of mains in a distribution system?

On the basis of pressure drop: The initial pressure and the minimum permissible terminal pressure are specified, and the pipe sizes are so chosen that the maximum estimated amount of steam may be transmitted without exceeding this pressure difference. The steam's velocity is disregarded and it may reach a magnitude in excess of 35,000 fpm which is not considered high.

2 ● a. What are the advantages and disadvantages of a low pressure distribution system?

b. High pressure?

a. The advantages of a low pressure distribution system include:

1. Smaller heat loss from the pipes.
2. Less trouble with traps and valves.
3. Simpler problems with pressure reducing equipment at the buildings.
4. No danger to building heating equipment from high pressure through failure of the reducing valves.

The disadvantages of a low pressure system are:

1. Larger pipe sizes.
2. Decreased field of usefulness owing to small pressure range.

b. The advantages of a high pressure system are:

1. Smaller pipe sizes.
2. Greater adaptability of the steam to various uses other than building heating.

The disadvantages of a high pressure system are:

1. Large heat loss from the pipes.
2. The high pressure traps and valves required often give more trouble than low pressure traps and valves do.
3. Extra heavy fittings are required.
4. Usually two reducing valves or some form of emergency relief is necessary to protect the building piping system.

3 ● Determine the size of pipe from the following data using Unwin's formula: Length of pipe, 600 ft.

Steam to be carried, 90,000 lb per hour, dry saturated.

Initial pressure, 100 lb per square inch, gage.

Final pressure, 40 lb per square inch, gage.

Using the formula:

$$P = \frac{0.0001306\ W^2L\left(1 + \frac{3.6}{D}\right)}{dD^5}$$

The pressure drop $P = 100 - 40 = 60$ lb per square inch.

The weight of steam per minute $W = \frac{90{,}000}{60} = 1500.$

The length of pipe in feet $L = 600$

The average density of steam d in pounds per cubic foot, taken from Keenan's Table:

$$\begin{aligned} \text{At 100-lb gage, } d &= 0.2578 \\ \text{At 40-lb gage, } d &= 0.1285 \\ \text{Average, } d &= 0.1932 \end{aligned}$$

The diameter of the pipe in inches $= D$.

Substituting the values in the formula:

$$60 = \frac{0.0001306 \times 1500^2 \times 600 \left(1 + \frac{3.6}{D}\right)}{0.1932 \times D^5}$$

$$D = 7.35 \text{ in.}$$

Therefore, an 8-in. pipe should be used.

4 ● What points should be borne in mind when laying out an underground steam conduit?

The conduit should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or the conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit.

5 ● What is considered the proper pressure for a hydrostatic test before completing the conduit?

In the case of any underground piping which is to be buried or otherwise made inaccessible, the assembled lines shall first be tested hydrostatically at a pressure of one and one-half times the maximum allowable service pressure and held for a period of at least two hours without evidence of leakage. In any case the hydrostatic pressure should not be less than 100 lb per square inch.

6 ● What factors should be considered before determining the route of a steam line?

1. The line should be so located that it will bring in the greatest revenue (or supply the most steam) with the least cost.
2. The ultimate length and size of services and branches necessary with each possible location should be estimated, for mains should be run near to the big loads.
3. The location of the boiler room or piping center of present and future buildings to be served should be considered.
4. Where possible, make the lines straight between manholes.
5. Avoid such obstructions as other lines, sewers, ducts, curb drains, manholes, valve boxes, catch basins, fire hydrants, and poles; especially avoid electric ducts and water lines.
6. Avoid locating lines near where pile driving and foundation construction for new buildings will take place.
7. Consider construction difficulties such as traffic, hard rock, and wet earth, which increase time and labor.
8. Consider the economies of using available sidewalk vaults of buildings. Weigh the advantage of less excavation against the cost of obstruction removal.
9. Consider all operating difficulties.
10. Consider the difficulties of negotiating agreements for lines on private property where public and private rights-of-way are available.
11. Consider the effect of proposed municipal and other improvements.
12. Consider municipal regulations.

7. ● State the advantages and disadvantages of tunnels over conduits.

The advantages of pipe tunnels over conduits are:

1. Accommodation for miscellaneous services other than steam.
2. Provision of an underground passage between buildings.
3. Easy installation of additional pipes and easy replacement of existing pipes with larger sizes.
4. Easy inspection and maintenance of pipes.

The disadvantages of pipe tunnels over conduits are:

1. Higher first cost.
2. Higher maintenance cost in general.

8 ● Is the steam consumption less in a building that shuts off its steam at night than in one that does not? Why?

It has been thoroughly demonstrated that the steam consumption is less in a building where the steam is shut off at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

9 ● Is the condensate from a building supplied with purchased steam always discharged to the sewer?

No. In some cities where the customers are not spread over too wide a territory and where natural water conditions make the treatment of boiler feed water expensive, the steam company provides mains for the return of the condensate to the boilers.

10 ● What are the common methods for salvaging heat in condensate?

The most common methods are:

1. The use of a water heating economizer for preheating the hot water supply to the building.
2. The use of a cooling radiator.

11 ● What are the common means used to graduate the heat supply according to variations in outside temperature?

a. A weather compensating thermostat regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply; at the same time it insures delivery of steam to all the radiators.

b. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds to produce some control over the heat output.

c. The use of an orifice system graduates heat supply.

d. The time-limit control which may be set to provide no service, continuous service, or periodic service, is also used. For periodic service, steam may be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. This type of service is provided by several intermittent settings. A night switch will maintain the intermittent day setting, or interrupt the day operation and cut off the supply of steam at night during any desired hours.

Chapter 43

WATER SUPPLY PIPING AND WATER HEATING

Maximum Possible Flow, Maximum Probable Flow, Average Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Hot Water Heating, Hot Water Storage, Swimming Pool Heating Requirements

DOMESTIC water supply systems present the engineer with a design problem that requires combining the somewhat empirical rules and formulæ in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same type of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulæ employed may have been derived on sound technical basis the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such *zoning*[1] or

[1]It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

sectionalizing is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions.

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

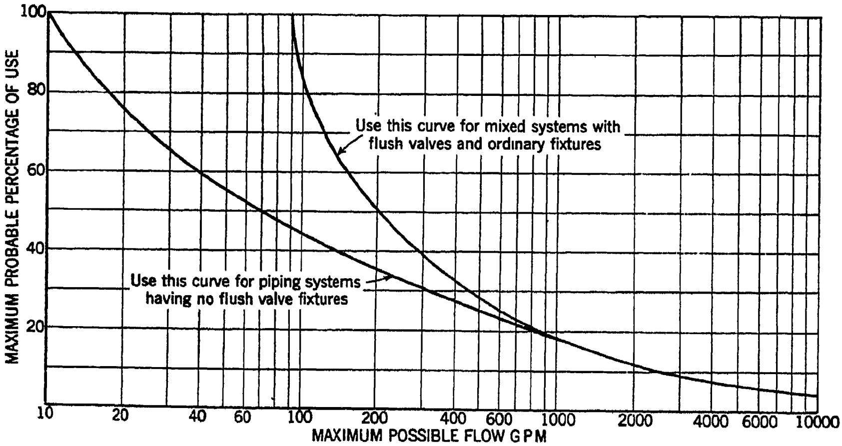


FIG. 1. CHART SHOWING RELATION BETWEEN MAXIMUM POSSIBLE FLOW AND MAXIMUM PROBABLE PERCENTAGE OF USE

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time*. Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal.

TABLE 1. APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

FIXTURES	COLD WATER (GALLONS PER MINUTE)	HOT WATER (GALLONS PER MINUTE)
Water-closets, flush valve	45[a]	0
Water-closets, flush tank	10	0
Urinals, flush valve	30[a]	0
Urinals, flush tank	10	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 4 in. heads, ½ in. inlets	3	3
Showers, 6 in. heads or larger	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	6	6
Wash trays	3	3
Laundry tray	6	6
Garden hose bibb	10	0

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 45 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety. Urinal flush valves would vary proportionately in the same manner.

In practice all the elements will vary according to conditions; in the case of flush valve closets the duration of flush with the kind and condition of supply apparatus, the interval between flushes with the number of people using the system and their habits; and the length of the rush period with the type of installation and its location. The effect of each of these time elements on the results should be considered in connection with any data on which it is based before passing judgment on the selection of the factor of usage. The longer the duration of the flush the greater is the probability of overlapping flow. In selecting the factor of usage shown in Fig. 1 for systems having flush valves, 10 seconds was chosen as the maximum duration of flush, a value that represents an approximate average as water closets are installed.

While the curve has been calculated for systems composed of water closets alone, it is possible to calculate probabilities for mixed systems of water closets and other smaller fixtures. It has been found however that for two systems both having the same maximum possible flow, one composed entirely of water closets and the other a mixed system of water

closets and smaller fixtures, the probability of a given rate of flow is greater for the system composed of water closets than for the mixed

Portion of Riser	Allowable Drop per Lb per 100 Ft	Maximum Probable Flow, Gallons per Minute: 5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	3.5	3/4	1	1 1/4	1 1/4	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2	2 1/2	3	3	3	3 1/2	3 1/2
S	20	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	2	2	2	2	2	2 1/2	3
R	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
Q	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
P	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
O	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
N	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
M	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
L	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
K	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
J	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
I	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
H	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
G	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
F	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
E	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
D	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
C	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
B	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2
A	30	3/4	3/4	3/4	1	1	1	1 1/4	1 1/4	1 1/4	1 1/2	1 1/2	1 1/2	2	2	2	2 1/2	2 1/2

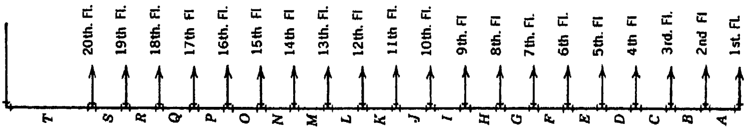


FIG. 2. TYPICAL RISER FOR 20-STORY BUILDING

system. The use of this chart then would produce results which would be on the safe side for mixed systems.

For systems composed entirely of fixtures other than flush valve fixtures the curve has been extended for smaller maximum possible flow values.

This chart applies to a normal building and not to installations where the inmates may all be required, for instance to bathe on certain days of the week and at certain hours of those days; or in schools for example where all the showers in the gymnasium may be used simultaneously after instruction periods. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water.

Cold Water

50 W. C. x 45 gpm	2250 gpm
50 Lavs. x 3 gpm	150 gpm
50 Sinks x 4 gpm	200 gpm
50 Baths x 5 gpm	250 gpm
Maximum possible flow	2850 gpm

Fig. 1 shows a factor of usage of 9 per cent.

Maximum probable flow of cold water is 2850 × 0.09 .... 257 gpm

Hot Water

50 Lavs. x 3 gpm	150 gpm
50 Sinks x 4 gpm	200 gpm
50 Baths x 5 gpm	250 gpm
Maximum possible flow	600 gpm

Fig. 1 shows a factor of usage of 23 per cent.

Maximum probable flow of hot water is 600 × 0.23 .... 138 gpm

Total for main supplying cold and hot water (2850 + 600) × 0.08 .... 276 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand.*

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used, and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantitities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than ¾ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43 × 100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30-lb drop under maximum probable demand is ample to take care of the friction caused by the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs; that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

TABLE 3. APPROXIMATE ALLOWANCES FOR FITTINGS AND VALVES IN FEET OF STRAIGHT PIPE

Size of Pipe (Inches)	Type of Fitting or Valve					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
½	4	3	8	2	48	8
¾	5	3	10	3	60	10
1	5	3	10	3	60	10
1¼	6	4	12	3	72	12
1½	7	5	14	4	84	14
2	7	5	14	4	84	14
2½	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

Where street water pressures are available the pressure drop through the meter and service pipe must be taken into consideration. Table 4 shows the pressure loss through meters. It also gives the minimum sizes of recommended service and maximum meter deliveries.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be 60 lb − (15 lb + 50 ft × 0.43 lb) = 60 lb − (15 lb + 21.5 lb) = 23.5 lb.

To change this into drop per 100 ft: $\frac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8$ lb per 100 ft.

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

TABLE 4. PRESSURE LOSS THROUGH WATER DISC METERS[a]

A. W. W. A. Standards

RATE OF FLOW GPM	APPROX. PRESSURE LOSS THROUGH METERS, LB PER SQ IN PIPE SIZE (IN.)							
	5/8	3/4	1	1½	2	3	4	6
5	1.5	0.5	0.2					
10	6.0	2.0	1.0	0.2				
15	14.0	5.0	2.0	0.6	0.2			
20	25.0	9.0	3.5	1.0	0.4			
25		13.5	5.5	1.5	0.6			
30		19.5	8.0	2.0	0.9			
35			11.0	3.0	1.0			
40			14.0	4.0	1.5			
45			18.0	5.0	2.0			
50			22.0	6.0	2.5	0.7		
75				14.0	5.5	1.5		
100				25.0	10.0	2.8	1.0	
125					15.0	4.0	1.5	
150					22.0	6.0	2.2	
175						8.0	3.0	
200						10.4	4.0	1.0
250							16.0	1.5
300							23.0	2.2
350								3.0
400								4.0
500								6.5
600								9.0
800								16.0
1000								25.0

MINIMUM SIZE OF SERVICE RECOMMENDED

RATE OF FLOW GPM	APPROX. MINIMUM PIPE SIZE OF SERVICE, MAIN TO METER (IN.) MAXIMUM LENGTH (FT)				
	30	75	100	150	200
1–20	3/4	3/4	1	1	1
20–30	3/4	1	1	1½	1½
30–50	1	1½	1½	1½	1½
50–100	1½	1½	2	2	2
100–150	1½	2	2	2½	2½

SAFE MAXIMUM DELIVERY OF METERS

METER SIZE IN.	CAPACITY, GPM BASED ON 25 LB LOSS THROUGH METER
5/8	20
3/4	34
1	53
1½	100
2	160
3	315
6	500
8	1000

[a]Pressure loss through compound and current meters are less than shown in table. For exact informati[on] consult manufacturers.

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), or flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1 \text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be 0.43 lb × 20 ft = 8.6 lb and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be 8.6 lb − 1 lb = 7.6 lb while the drop per 100 ft of equivalent run will have to be $\frac{1 \text{ lb} \times 100}{600} = 0.1667 \text{ lb}$.

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8-in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

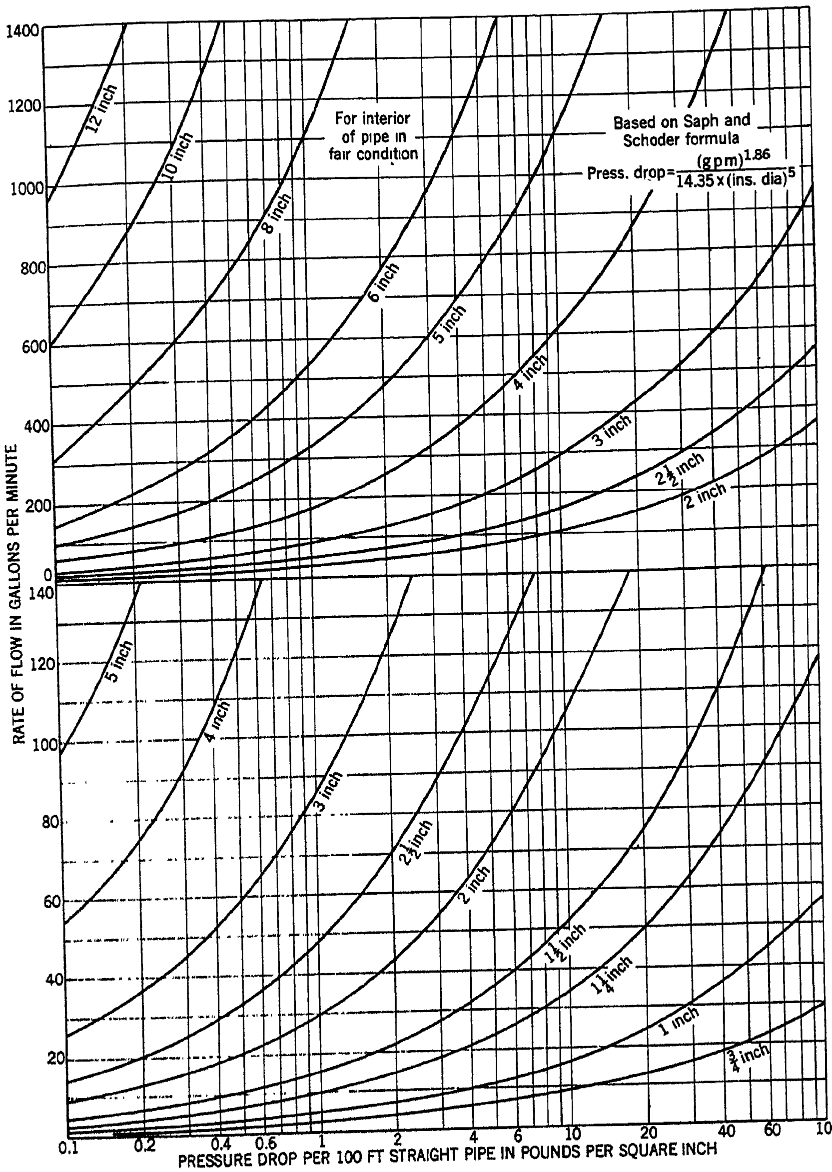
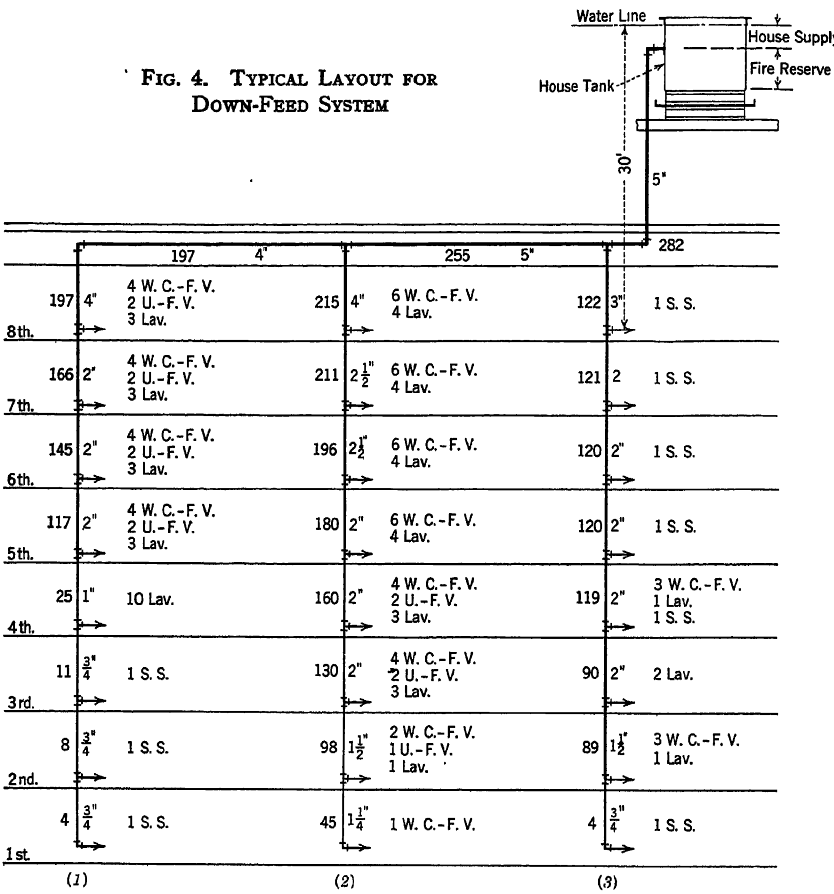


FIG. 3. CHART GIVING PRESSURE DROP FOR VARIOUS RATES OF FLOW OF WATER

still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of 1 lb + 4.3 lb = 5.3 lb which, divided by the 600 ft equivalent run gives a drop per 100 ft of

$$\frac{5.3 \times 100}{600} = 0.9 \text{ lb}$$

FIG. 4. TYPICAL LAYOUT FOR DOWN-FEED SYSTEM



and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 5 in., and 4 in., respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be $\frac{5.3 \text{ lb} \times 100}{200} = 2.7$ lb per 100 ft. This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

TABLE 5. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 1. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	1 S. S.	4	4	8	100	8	30	¾
3rd	1 S. S.	4	4	12	92	11	30	¾
4th	10 Lav.	3	30	42	58	25	30	1
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	291	40	117	30	2
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	540	27	145	30	2
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	1038	19	197	2	4

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 5, 6, 7 and 8 show the schedule for Risers Nos. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 9 and 10. This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 11 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up 0.43 lb × 80 = 34.4 lb and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as an even 300 ft.

TABLE 6. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 2. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 W. C.	45	45	45	100	45	30	1¼
2nd	2 W. C. 1 U. 1 Lav.	45 30 3	90 30 3 123	168	58	98	30	1½
3rd	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	417	31	130	30	2
4th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	666	24	160	30	2
5th	6 W. C. 4 Lav.	45 3	270 12 282	948	19	180	30	2
6th	6 W. C. 4 Lav.	45 3	270 12 282	1230	16	196	30	2½
7th	6 W. C. 4 Lav.	45 3	270 12 282	1512	14	211	30	2½
8th	6 W. C. 4 Lav.	45 3	270 12 282	1794	12	215	2	4

Then the allowable drop per 100 ft will be $\frac{25.6 \text{ lb} \times 100}{300} = 8.5$ lb and the sizes shown in Fig. 5 are based on this amount of drop. Of course the other risers will have the same maximum flows at the bottom as they formerly had at the top, namely 215 and 122 gal, respectively, for Risers Nos. 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 255 gpm, and between Riser No. 3 and the street main, 282 gpm which at a drop of 8.5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also because many plumbing fixtures have no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably

TABLE 7. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 3. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	3 W. C. 1 Lav.	45 3	135 3 138	142	63	89	30	1½
3rd	2 Lav.	3	6	148	61	90	30	1½
4th	3 W. C. 1 Lav. 1 S. S.	45 3 4	135 3 4 142	290	41	119	30	2
5th	1 S. S.	4	4	294	41	120	30	2
6th	1 S. S.	4	4	298	40	120	30	2
7th	1 S. S.	4	4	302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	2	3

required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follows:

1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.
2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed.
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

TABLE 8. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

RISER No.	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN.
1	1038	1038	18	187	2	4
2	1794	2832	9	255	2	4
3	306	3138	9	282	2	5

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve, set on a 45-deg angle, and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 6.

TABLE 9. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISERS WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY (SEE FIG. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
	Riser No. 1							
7th and below				789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	10 10 3	40 20 9 69	858	20	172	3.3	4
	Riser No. 2							
7th and below				1512	14	211	30	2½
8th	6 W. C. 4 Lav.	10 3	60 12 82	1594	14	223	3.3	4
	Riser No. 3							
7th and below				302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	3.3	3

For the second arrangement of hot water risers (Fig. 7*b*), circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across on to the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement (Fig. 7*c*) all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the

TABLE 10. SUMMARY OF RISER SIZES TO GIVEN MAIN SIZES WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY. (SEE FIG. 4)

RISER No.	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN.
1	858	858	20	172	3.3	4
2	1594	2452	10	245	3.3	4
3	306	2758	9	248	3.3	4

regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the bottom of each riser should be arranged in the manner already outlined and any riser not going to the basement to supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

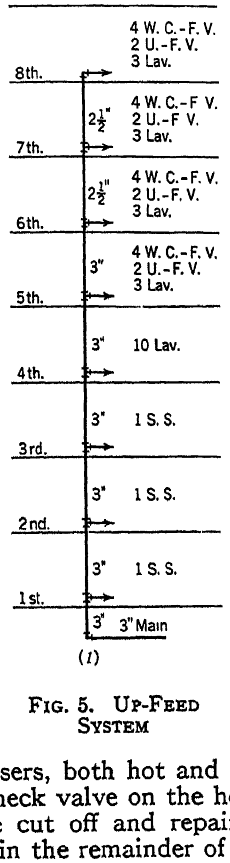


FIG. 5. UP-FEED SYSTEM

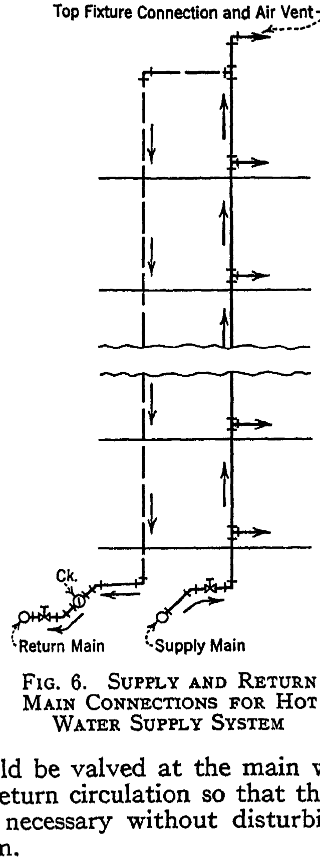


FIG. 6. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the

TABLE 11. TYPICAL CALCULATION OF PIPE SIZES ON UP-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS (SEE FIG. 5)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	249	44	109	8.5	2½
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	498	28	139	8.5	2½
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	747	22	164	8.5	3
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	996	18	179	8.5	3
4th	10 Lav.	3	30	1026	18	185	8.5	3
3rd	1 S. S.	4	4	1030	18	186	8.5	3
2nd	1 S. S.	4	4	1034	18	187	8.5	3
1st	1 S. S.	4	4	1038	18	188	8.5	3

TABLE 12. SUGGESTED STORAGE TANK SIZES FOR HOMES AND APARTMENTS

ALL YEAR SERVICE BASED ON BOILER WATER AT 180 F				SERVICE DURING HEATING SEASON BASED ON BOILER WATER AT 215 F			
Tank Capacity Gal	Piping Connections		Number of Baths or Families	Tank Capacity Gal	Piping Connections		Number of Baths or Families
	Boiler, In.	Tank, In.			Boiler, In.	Tank, In.	
30	1	¾	1	30	1	¾	1
35	1¼	¾	1	40	1	¾	1
40	1¼	¾	1–2	52	1	¾	1
50	1¼	¾	1–2	66	1¼	1	1–2
60	1¼	1	1–2	82	1¼	1	2–3
72	1½	1	2–3	100	1¼	1	3
80	2	1	2–3	120	1¼	1	4
100	2	1¼	3–4	144	1½	1	5
125	2	1¼	4–5	160	2	1¼	6
150	2	1¼	5–6	200	2	1¼	6–7
200	2	1½	6–7	250	2	1½	7–9
250	2½	1½	7–9	300	2	1½	9–11
300	2½	1½	9–11	400	2½	2	11–15
400	3	2	11–15	500	2½	2	15–18
500	3	2	15–18	600	3	2½	18–21

water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day is assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is

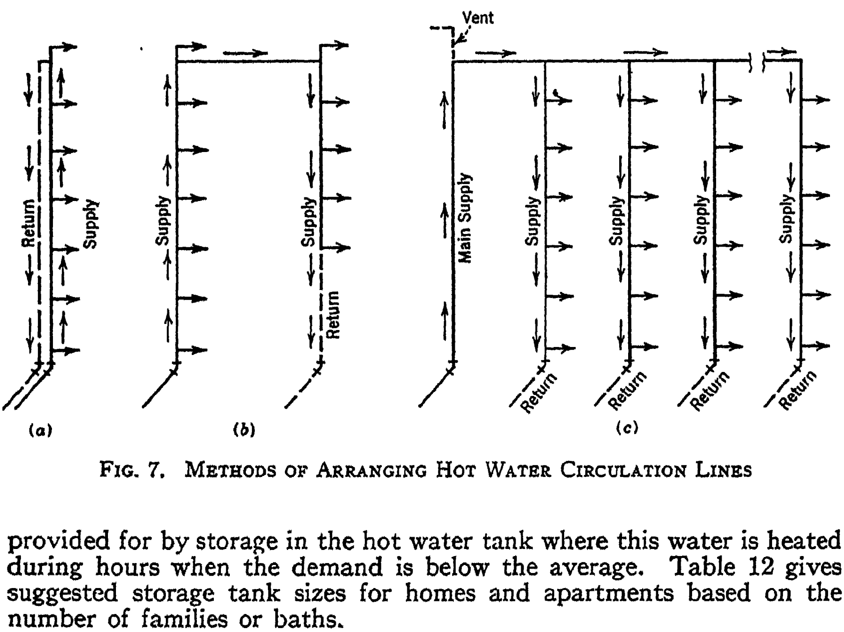


FIG. 7. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average. Table 12 gives suggested storage tank sizes for homes and apartments based on the number of families or baths.

HOT WATER HEATERS

Various types of heaters are available for supplying the hot water for domestic service in buildings. In any hot water supply system the water should be heated to a temperature between 150 and 180 F. Where the hot water requirements include supplies for kitchens, laundries or process work, the higher temperatures are used. In buildings where steam is available throughout the year, the hot water supply is usually taken from this source. In smaller domestic installations the fuel-burning device is generally automatically arranged so that hot water is supplied the entire year and not merely when the boiler is used for heating purposes.

Water is heated by various methods using heat exchangers arranged so that the boiler heating medium gives up its heat to the water in the hot water circulating system. These heat exchangers may be classified as follows:

1. Submerged steam heating coil in storage tank.
2. Submerged water heating coil in storage tank. (Fig. 8).
3. Indirect water heater, mounted on side of boiler below water line. (Fig. 9).
4. Submerged indirect water heater, placed in boiler below water line. (Fig. 10).

The efficiency of these heaters may be estimated as nearly 100 per cent as the heat loss from surface radiation of the heater and tank shell when covered with insulating material is generally reduced to a minimum. The capacities of these heaters are usually available from manufacturers

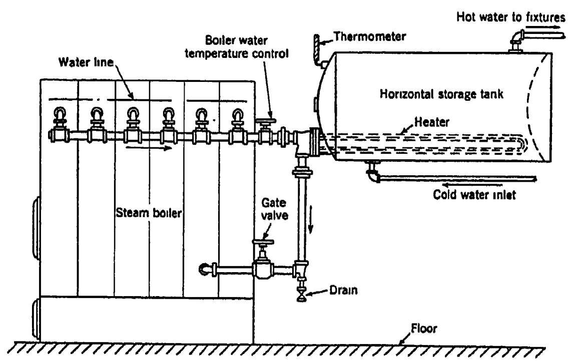


FIG. 8. HOT WATER HEATING COIL SUBMERGED IN STORAGE TANK

rating tables. The area of the inside surface of a heating coil may be determined from the following equation:

$$A = \frac{Q \times 8.33\ (t_o - t_i)}{K_o \times t_m} \tag{1}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_o = hot water outlet temperature, degrees Fahrenheit.
t_i = cold water inlet temperature, degrees Fahrenheit.
K_o = coefficient of heat transmission, Btu per hour per square foot surface.
For copper or brass coils K_o = 240 (steam) and 100 (hot water).
For iron coils K_o = 160 (steam) and 67 (hot water).
t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature. t_m is approximately = $\left[t_s - \frac{(t_o + t_i)}{2}\right]$

Equation 1 may also be used for determining revised heating coil ratings under different temperature conditions as stated in the manufacturers ratings. When selecting a water heater, the conditions of operation should be carefully considered, as well as the location of the

storage tank and the piping arrangement between the boiler, heater and tank. It is generally good practice to allow a margin of safety when selecting an indirect heater of the proper size to provide for loss of efficiency due to the accumulation of scaling on the coils and piping. Heat exchangers classified according to (3) and (4) may be used with or without a storage tank, but when tanks are omitted, the indirect water heaters should be increased in size so as to heat the water instantaneously as it is needed.

The storage tank should be installed as high as possible. Horizontal tanks are preferable for all medium size installations and absolutely essential on larger installations. Where possible the storage tank should be installed with the bottom of the tank at or above the boiler water line. Horizontal storage tanks smaller than 18 or 20 in. diameter are not

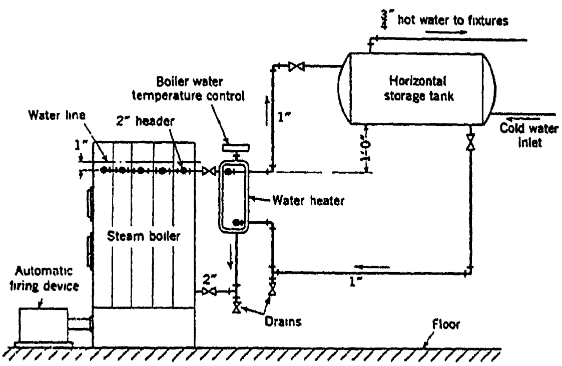


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

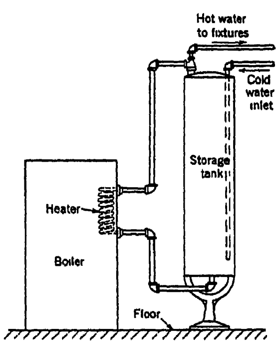


FIG. 10. INDIRECT WATER HEATER PLACED IN BOILER

recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn.

Pipe sizes between the water heater and boiler should be full size of the heater tappings (Table 12). When a heater is connected to a horizontal sectional boiler, it is recommended that connections be made to all sections and joined together a few inches below the water line as shown in Fig. 8, so that steaming is prevented in those sections which are not connected to the header.

When a steam coil is used for heating the water, an automatic thermostatic valve may be installed in the steam supply to the coil. The operation of this automatic valve is controlled by a thermostat located in the storage tank which permits the proper amount of steam to enter the coil so as to maintain an even water temperature.

An indirect water heater may be used on either a steam or hot water system, and generally this type of heater is provided with a temperature control device located in the boiler water circulating connection to the water heater. The setting on this thermostatic valve may be as low as

140 F or as high as 180 F and may be readily adjusted to meet particular requirements. With this type of control it is impossible to overheat the hot water supply which is an important safety consideration in some installations. This type of system may also be conveniently used during the non-heating season with the operation of the fuel burning device controlled by the water heater thermostat. (See Chapter 37). During the heating season the water heater temperature control functions as a low limit control.

When an indirect water heater is applied to a gravity hot water system, it is necessary to provide a valve in the supply to the heating system to prevent the flow of hot water from the boiler when heat is not required in the house. This valve may be controlled from a room thermostat and the automatic fuel-burning device controlled from the water heater thermostat. To prevent circulation in a forced hot water heating system flow control valves may be installed in the flow and return lines which act merely as check valves when the circulating pump is not operating. In this arrangement the pump is controlled by the room thermostat and the automatic fuel-burning device is controlled from the water heater thermostat.

STORAGE CAPACITY AND BOILER ALLOWANCES

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about 500×40 gal = 20,000 gal and this is an average of $\frac{20{,}000 \text{ gal}}{24} = 833$ gph but the peak hour will require $\frac{1}{10}$ of 20,000 = 2000 gal and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be $2000 - 833 = 1167$ gal so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be $\frac{1167}{0.75} = 1556$ gal.

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to say 1200 gph which then gives $2000 - 1200 = 800$ gal as the shortage during the peak hour, and the necessary storage will be $\frac{800 \text{ gal}}{0.75} = 1067$ gal; or the heating capacity can be increased to 1500 gal, leaving a shortage of $2000 - 1500 = 500$ gal.

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage, as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 4 sq ft of equivalent steam heating surface[3] (radiation) must be provided for every gallon of water heated 100 F or from 50 F to 150 F, which is the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (gph)	Additional Boiler Capacity (Sq Ft EDR)
833	3332
1200	4800
1500	6000

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

The boiler allowance value of 4 sq ft of equivalent steam radiation for each gallon of water heated through a temperature range of 100 F is based on an hourly heating rate. When reduced heating capacities are desired for economic reasons of boiler design and selection, engineers frequently recommend that the heating rate be extended over a period of two hours in which case the boiler allowance value would be reduced to 2 sq ft of equivalent steam radiation. Similarly any other heating rate may be established and a corresponding value of boiler allowance determined.

Reliable information based upon the installations of several heaters in existing heating systems indicates varying arbitrary values of boiler allowances to be used. When these values are selected for usage, a careful analysis of the varying factors involved in determining these values should be considered so that the proper heating allowances may be provided.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 13 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 14 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

[3]Actual requirement for 100-deg temperature difference = $\frac{100 \times 8.33}{240}$ = 3.33 sq ft per gallon of water heated.

20 Baths at 40 gal and 33 per cent	270 gal
20 Lavs. at 20 gal and 25 per cent	100 gal
20 Sinks at 30 gal and 33 per cent	200 gal
20 Trays at 50 gal and 60 per cent	600 gal
Total	1170 gal
Probable peak use at one time	35 per cent
Probable actual peak demand	409 gph

If three persons are assumed to an apartment the total daily use of hot water should approximate 20 × 3 × 40 gal = 2400 gal and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal, which indicates that the values in Table 13 are safe.

SWIMMING POOL HEATING REQUIREMENTS

Swimming pools present a problem of hot water heating demand which is frequently overestimated. Few outdoor swimming pools require water heating, and in some cases they require the addition of cold water to regulate the temperature. The recirculation system of a swimming pool consists of the pumps, hair and lint catchers, and filters together with all necessary pipe connections to the inlets and outlets of the pool. The water heater, the sterilizing equipment and suction cleaner are usually installed or connected to the recirculation system and may be considered as integral parts of the system.

The recirculation system and all its component parts should be designed to provide the required volume of circulation so that the water turnover ratio is at least two times per day and where heavy loads are anticipated the turnover ratio should be increased to three times or more. Many states have regulations prescribing the circulation turnover.

The water heaters for swimming pools are usually instantaneous steam

TABLE 13. ORDINARY MAXIMUM HOURLY DEMAND FOR HOT WATER FOR VARIOUS FIXTURES IN GALLONS AND PROBABLE PERCENTAGE OF USAGE

Type of Building	Lavatories		Baths	Showers	Slop Sinks	Kitchen Sinks	Pantry Sinks	Foot Baths	Wash Trays	Av. Max. Use[a]
	Private	Public								
Maximum Probable Usage GPH	20	20	40	300	30	30	20	20	50	
	Probable Usage in Per Cent of Maximum Ordinary Use									
Apt. house	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gym.	25	100	100	100	----	----	----	100	----	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	----	100	----	90
Laundries	25	100	----	----	33	----	----	----	100	100
Office building	25	75	----	----	50	----	----	----	----	20
Baths	25	150	150	100	50	----	----	----	----	100
Residences	25	----	50	33	50	33	50	50	60	50
Schools	25	75	----	100	67	33	100	50	----	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

[a]Percentage of fixtures likely to be demanding maximum probable usage at any one time.

coil heaters. These heaters should be sized so that they will have sufficient capacity to heat the water delivered by the circulating pump 15 F per hour.

The water temperature in a pool is usually maintained at about 72 F. A few states have regulations prohibiting higher water temperatures than 70 F. The room temperature should be approximately 5 F higher, but not more than 8 F higher nor less than 2 F lower, than the water temperature.

Example 6. Assume a swimming pool 75 ft long, 30 ft wide with an average depth of 6 ft. If the water is to be heated from a temperature of 50 to 65 F, what capacity heater and steam consumption is required with a turnover ratio of two times per day?

Pool volume: $75 \times 30 \times 6 \times 7.5 = 100{,}000$ gal.

With a turnover ratio of twice in 24 hr, the heating capacity is: $\frac{100{,}000 \times 2}{24} = 8333$ gal per hour.

The steam consumption would be: $\frac{8333 \times 8.33\ (65 - 50)}{970} = 1080$ lb steam per hour.

Regulation of swimming pool temperatures is essential for successful operation and economy. It is therefore recommended that the steam supply to the heater be provided with a by-pass which may be used for pool filling and initial heating and that a smaller by-pass be installed with an automatic control valve having the capacity to heat the circulation water approximately 5 F per hour.

TABLE 14. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	
	(Transient)	40 (per day)
	(Men)	40 (per day)
	(Mixed)	60 (per day)
	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture)
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurants	\$0.50 Meals	0.5 (per customer with hand washing)
		1.0 (per customer with machine washing)
	\$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	\$1.50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

PROBLEMS IN PRACTICE

1 ● The heating capacity of an indirect water heater is 100 gal per hour, using steam at 215 F and raising the water from a temperature of 50 to 150 F. Determine the heating capacity of the same water heater using water at a temperature of 180 F for the heating medium.

Using Equation 1, and because the surface area of the water heater is the same for each condition, the two conditions may be equated as follows:

$$\frac{100 \times 8.33\ (150 - 50)}{240\left[215 - \dfrac{(150 + 50)}{2}\right]} = \frac{Q \times 8.33\ (150 - 50)}{100\left[180 - \dfrac{(150 + 50)}{2}\right]}$$

Q = 28.98 gal per hour, capacity of heater using water at a temperature of 180 F.

2 ● Why is it impractical to size water supply piping so pipe friction will produce an equal pressure on each fixture?

Because the friction would be built up only in periods of maximum flow and at all other times it would be only a fraction of that required.

3 ● What is the purpose of zoning water supply systems in tall buildings?

To avoid excessive pressures in the lower stories.

4 ● Define the maximum possible flow, the maximum probable flow, and the average probable flow.

The maximum possible flow is the flow which would occur if all of the outlets on the system were opened at one and the same time. The maximum probable flow is the flow which will occur with probable peak conditions. The average probable flow is the flow likely to occur under a normal condition of use.

5 ● What is the factor of usage?

This is the percentage of the maximum possible flow which is likely to occur at peak load.

6 ● How many feet higher than the uppermost fixtures must the water line in a house tank be to provide about 15 lb per square inch pressure at the fixture outlet?

Allowing for pipe losses, about 45 ft.

7 ● What methods of hot water circulation commonly are employed with hot water supply systems?

a. Up-feed risers with returns having no connections paralleling the risers.
b. Up-feed risers with returns in other locations, and with connections taken off both supply and return.
c. One main up-feed riser, without connections, supplying all down-feed risers for all fixtures.

8 ● Which method of hot water supply generally is the most satisfactory?

The single main up-feed riser supplying drop risers for all fixtures.

9 ● How much of the water stored in a hot water storage tank really is available for use?

About 75 per cent, because when only 25 per cent of the original water remains in the tank it has been so cooled down by the entering water that it is too cold for satisfactory use.

10 ● In cases of intermittent demand, does a large hot water storage tank increase or decrease the steam load for water heating?

It decreases the steam load in cases of intermittent demand but causes no change in the steam load if the demand is constant.

Chapter 44

TEST METHODS AND INSTRUMENTS

Pressure Measurement, Temperature Measurement, Air Movement, Humidity Measurement, Carbon Dioxide Determination, Dust Determination, Flue Gas Analysis, Measurement of Smoke Density, Heat Transmission, Eupatheoscope

SEVERAL types of measuring apparatus are available for accur determining the thermal capacity and air movement of gaseous v and homogeneous materials. This chapter gives a brief description (principal instruments used in connection with the proper contro testing of heating and air conditioning installations.

TEST METHODS

The SOCIETY has adopted standard test methods or codes for t and rating most heating, ventilating and air conditioning equipment. A list of the titles of these test codes may be referred to on pages 839 and 840. Many of the test instruments required are specified and described in these codes.

PRESSURE MEASUREMENT

Atmospheric pressure is usually measured by a *mercurial barometer* which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The pressure of the atmosphere on the exposed top of the mercury in the cistern supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern. Atmospheric pressure is the same as the pressure exerted by this supported column of mercury, and, in pounds per square inch, is equal to its height in inches times 0.491, which is the weight in pounds of 1 cu in. of mercury at 32 F. At latitude 45 deg and sea level, and at a temperature of 32 F, the atmosphere will support a column of mercury 29.921 in. in height. The pressure of 14.7 lb per square inch, derived by multiplying 29.921 by 0.491, is called *standard* or *normal barometric pressure.* Since the height of the barometer depends on the density of the mercury as well as on the pressure of the atmosphere, and since the density is dependent on the temperature, mercurial barometer readings should always be corrected for temperature.

The following equation may be used to make corrections for temperature:

$$h = h_1 [1 - 0.000101 (t_1 - t)] \quad (1)$$

where

h = height of mercury column corrected to temperature t, inches.
h_1 = actual height of mercury column, inches.
t_1 = actual temperature of mercury column, degrees Fahrenheit.
t = temperature to which column is to be corrected, degrees Fahrenheit.

Atmospheric pressure may also be measured by means of an *aneroid* barometer. In this instrument atmospheric pressure is made to move an indicating pointer either by bending the thin corrugated top of a partially exhausted metallic box, or by distorting a bent, thin-walled metal tube. The aneroid barometer contains no liquids, is portable but is less accurate than the mercurial barometer.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage*, and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage*. The most common type of these gages contains a flexible hollow metal tube of oval cross section, known as a *Bourdon tube*. When subjected to unequal inside and outside pressures, this tube tends to straighten out, and a pointer motivated by this straightening indicates the pressure difference on a suitable graduated scale.

High vacuum readings such as are encountered in condenser and steam jet refrigeration practice are commonly obtained by the use of mercury column vacuum gages. When the readings obtained with the mercurial barometer and those with the mercury vacuum gage have both been corrected to 32 F, the difference in the two readings will give the absolute vacuum in inches of mercury. Equation 1 may be used to make corrections for temperature.

In the measurement of small pressure differences, the *U* tube in one of its many forms is convenient, inexpensive and it may be built for any desired degree of accuracy. *U* tube manometers may be fabricated from glass and rubber tubing or any of the numerous commerical forms may be used.

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage*. It is essentially a *U* tube containing either water, kerosene, alcohol, or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pressure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

Various forms of high sensitivity draft gages[1] frequently called micromanometers[2] are available for the measurement of small pressure differen-

[1]Fluid Velocity and Pressure, by J. R. Pannell (*Edward Arnold and Co.*, London, 1924).
[2]Illinois Micromanometer, University of Illinois (*Engineering Experiment Station Bulletin* No. 120, p. 91).

tials and may be sensitive to pressures as small as 0.001 in. of water. These gages are often useful where measurements are to be made on pressure differentials less than 0.1 in. of water, although their total range may extend as high as 5 to 10 in. of water.

TEMPERATURE MEASUREMENT

In engineering work, thermometers are largely employed to measure the intensity of heat. Those most commonly used are *liquid-in-glass* thermometers. Mercury and alcohol are the liquids most frequently used.

Mercurial thermometers depend on the uniform expansion of mercury to indicate changes in temperature. An amount of mercury held in a sealed tube with a bulb at one end will rise to one definite level when immersed in melting ice, and to another definite level when immersed in boiling water. These two points are marked, and the space between them is divided into a number of equal portions, each of which is called a degree. In the Fahrenheit scale, there are 180 deg thus obtained, while the centigrade scale has 100 and the Reáumur has 80. Like divisions are marked off on the column above and below these two determined points in order that a greater range of temperature may be read.

Mercurial thermometers may be used in a temperature range from -40 to $+932$ F.

Alcohol thermometers are similar in construction to mercurial thermometers but are useful in a lower temperature range (-94 to $+248$ F).

Industrial thermometers in a large number of designs are available, but for test purposes etched stem thermometers are most frequently used. The etched stem thermometer has greater sensitivity and less lag than most industrial thermometers.

For precision temperature measurements, it is necessary to correct the thermometer reading for emergence of the stem if any part of the mercury column is exposed to a temperature other than that being measured (unless the thermometer has been calibrated under like conditions). The emergent stem correction may be calculated by the following equation:

$$K = 0.00009\, D\, (t_1 - t_2). \tag{2}$$

where

K = correction to be added, degrees Fahrenheit.
D = length of emergent stem, degrees Fahrenheit on thermometer stem.
t_1 = temperature indicated on the thermometer, degrees Fahrenheit.
t_2 = temperature of exposed mercury stem, degrees Fahrenheit.

Thermocouples[3] may be used to measure any range of temperatures up to 2,900 F. When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends on the character of the metals and the difference in temperature between the junctions. A potentiometer or sensitive galvanometer of high resistance connected to the thermocouple will give a deflection which is a function of the temperature difference between the hot and cold junctions. Thermocouples con-

[3]Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 55).

nected in series are called *thermopiles*. Thermocouples for the measurement of high temperatures are calibrated with the aid of the known melting points of pure metals.

Resistance thermometers are suitable for temperature measurements up to 1800 F. These thermometers depend for their operation on the change of resistance with temperature of a platinum, nickel, or copper wire coil, and they are calibrated in the same way as thermocouples.

Pyrometers of various types may be used for temperatures above 500 F. The *mercurial pyrometer* is a thermometer with an inert gas, such as nitrogen or carbon dioxide, above the mercury column to prevent the mercury from boiling. The *radiation pyrometer* consists of a thermopile upon which the radiation from a hot source is focused by a concave mirror or lens. A sensitive galvanometer or potentiometer with a calibrated temperature scale indicates the thermo-electromotive force created by the heat on the thermopile. The *optical pyrometer* measures radiant energy by comparing the intensity of a narrow spectral band, usually red light emitted by the object, with that emitted by a standard light source (electric lamp). *Thermo-electric pyrometers* operate on the same principle as thermocouples. When measuring high temperatures, it is customary to hold the cold junction at room temperature and this may cause some error if the room temperature is above or below the calibration point. For extremely precise temperature measurements, the cold junction is usually immersed in melting ice to fix the cold junction temperature. Various forms of hand-operated and automatic cold junction temperature compensators are also available.

In the measuring of room temperature care must be exercised to prevent the results from being affected by the body heat of the observer, by air currents from doors, windows and other openings, or by radiant heat from some local source such as a radiator or wall. All glass thermometers should be mercury thermometers with engraved stems. The total graduations of the thermometers should be from 20 to 120 F, in one degree graduations. No ten degrees should occupy a space of less than one-half inch. The accuracy throughout the whole scale must be within one-half degree. The operator should take hold of the top and no part of the body, including the hand, should be nearer than 10 in. to the bulb. The thermometer should not be closer than 5 ft to any door, window, or other opening; should not be closer than 12 in, to any wall; and should be between 3 and 5 ft from the floor. A sling instrument should be used for extreme accuracy. Thermocouples or resistance thermometers may also be used for room temperature measurements, an advantage being that the operator can read temperatures from outside the room if desired, and thus eliminate the errors which might be caused by his presence close to the temperature measuring device.

For measuring duct temperatures a duct thermometer should be used, with the bulb extending into the duct at least 6 in. When the thermometer is to be permanently located in the duct, a pipe flange or nipple should be used to receive the threaded portion of the thermometer stem. When the thermometer is not to be permanently located, a cork or rubber stopper may be placed around the stem to prevent errors from air leakage. Readings should be taken at various locations in a duct so due consideration may be given to temperature stratification. Other forms of

temperature measuring devices may be used, but the active part must be at least 6 in. from the duct wall.

Recording instruments may be used for testing and for making continuous records of operation. Potentiometer and Wheatstone bridge recorders for thermocouples and resistance thermometers respectively may have accuracies of ± ⅓ per cent of their range, or, for example, to ± 1 F in a range of 0 to 300 F. This accuracy compares favorably with that of other forms of temperature measuring devices.

AIR MOVEMENT MEASUREMENT

The quantity, velocity and pressure of air moved by a fan or flowing through a duct or grille may be determined by various methods. The instruments in common use are the Pitot tube, anemometer, direct reading velocity meter, and Kata-thermometer, the latter being suitable for low air velocities and being commonly used for measurements at points where the air is not confined in a duct. Electrical anemometers are also available, operating on the principle of measurement of the variation of resistance of a hot wire cooled to various degrees by air velocities past the wire. The use of calibrated nozzles, orifice plates, and Venturi meters are recognized methods, which, however, have little application in connection with ventilation practice.

Pitot Tube

This usually consists of two tubes, one within the other, which when properly held in the air stream will register the total or impact pressure and the static pressure, respectively. If these tubes are connected to opposite sides of a draft gage, or other type of *U* tube, the recorded pressure will be the differential or velocity pressure. Volume measurements may thus be made in a duct of known area. Pitot tube measurements are preferably used for air velocities exceeding 20 fps. Volumetric determinations from Pitot tube readings should take into account the barometric pressure and the temperature and humidity of the air measured.

Air flow in ventilation practice is generally in the turbulent range. When stratification of velocity, vortex motion, or violent eddy currents of air in ducts exist, accurate velocity pressure measurements are difficult. To insure accuracy a straight section of duct from 5 to 10 times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work it is necessary to make a traverse of the duct, dividing its cross section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

A pitot tube of standard design and the traverse method of obtaining average velocity are completely described in the A.S.H.V.E. Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers.[4]

For precise work the shape, size and calibration of the pitot tube are important considerations in the determination of the correct air flow.

[4]A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407. Amended June, 1931.

Extensive test results comparing the characteristics of several pitot tube types are available in the published reports[5] of the government.

Anemometer

The vane-type anemometer is most frequently used for test work. It consists of a small, delicate, fan-like rotor connected to a revolution counter. The instrument is held in the air stream where the velocity is to be measured. It is calibrated to read directly in linear feet. The velocity in feet per minute is obtained by dividing the reading (linear feet) by the elapsed time, in minutes.

The vane anemometer is delicate, requires frequent calibration and is suited only to low velocities (less than 3000 fpm). The vanes of the instrument should never be touched.

The following procedure for obtaining anemometer readings is based on research conducted at *Armour Institute of Technology* in cooperation with the A.S.H.V.E. Research Laboratory[6].

Supply Grilles. The surface of the grille should be marked off into a number of equal areas approximately 6 in. square. A 4-in. anemometer should be used and should be held at the center of each section in contact with the grille (or as close as possible) for a period of time sufficient to insure an average reading. In the case of supply grilles, the instrument should always be held with the dial facing the operator. The average of the corrected readings should then be used in the following formula to obtain the flow in cubic feet per minute:

$$cfm = CV\frac{A + a}{2} \text{ or } \frac{CVA\,(1 + p)}{2} \tag{3}$$

where

V = average of corrected anemometer readings, feet per minute.

A = gross area of grille, square feet.

a = net free area of grille, square feet.

p = percentage of free area of grille expressed as a decimal.

C = a coefficient that varies with the velocity from grille and may vary slightly with type of grille. For average use, with supply grilles, C can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities.

Particular care should be exercised in the case of long, narrow grilles. The nature of the approach sometimes results in there being a narrow strip along the top or bottom of the grille through which no air will be flowing. This may be detected by holding the anemometer completely out of the air stream and then moving it slowly inward over the grille until the vanes just start to move. The distance which the vanes extend over the grille opening at this moment will indicate the width of the dead strip. Only the remaining portion of the grille should be considered in making the calculations for gross and free area.

Exhaust Grilles. The surface of the grille should be marked off and readings taken in the same manner as with supply grilles, except that the instrument should be held with the dial facing the grille, and in contact with it. The traverse should be taken at a uniform rate, allowing suf-

[5] Technical Notes No. 546, *National Advisory Committee for Aeronautics*, November, 1935.

[6] Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 201, Vol. 37, 1931, p. 619, and Vol. 39, 1933, p. 373).

ficient time in each space to minimize the percentage of error. In the case of exhaust grilles it is found that the formula:

$$cfm = KVA \tag{4}$$

in which

V = average indicated velocity obtained by the anemometer traverse.
A = gross area of grille, square feet.
K = coefficient determined by experiment. For average use, with exhaust grilles, K may be taken as 0.8 for all usual velocities.

This formula is of advantage, especially with ornamental grilles, in that the free area need not be measured.

The flow of air through registers and grilles is of considerable importance, being frequently the only convenient method of measuring the volume of supply air to a room. While duct measurements, if available, are more dependable, grille measurements provide a fairly accurate method, if care is taken in the technique of using the anemometer.

Direct Reading Velocity Meter

An instantaneous direct reading air velocity instrument available in a portable case is used for recording air movement on a calibrated scale. Air entering the meter actuates a vane movement to which is attached a pointer with control hair springs and a magnetic damping arrangement.

Velocity meters are available in either orifice, shutter or tube types. The orifice unit is used where the instrument can be placed in the air stream when obtaining a reading such as in rooms or large spaces or at unrestricted outlets of ducts. The use of the shutter type is similar to the orifice style except that it has means for changing the scale range. The shutter is adjusted so that the large ports are fully open for low velocity readings. For high range readings the shutter is turned until the large openings are closed and only a small port is open. The shutter is omitted in the tube type of meter and in place of this fitting a tube attachment is threaded to the case. A flexible rubber tube and specially designed metal jets are used for obtaining high range readings. Jets may be secured for unusual applications such as in obscure locations, surging air currents or leakage from ducts and similar requirements. Due to the connecting tube flexibility, the jet can be moved as required while the instrument is held stationary.

Where it is desired to obtain air velocity readings within a duct, special jet and additional meter fittings are used which indicate directly the true air velocity with no corrections being essential for static pressure conditions. Air enters the meter through one side of the jet and is discharged back into the duct through the other side of the jet.

Kata-Thermometer

The Kata-thermometer can be used to determine air velocities provided the walls and surrounding objects are at or near the room temperature. Especially at low velocities it constitutes a useful instrument for readily detecting drafts.

The instrument is essentially an alcohol thermometer with a bulb approximately 5/8 in. in diameter and 1/2 in. long with a stem 8 in. long reading from 100 F to 95 F, graduated to tenths of a degree. To take readings the bulb is heated in water until the alcohol expands and rises into a top reservoir. The time in seconds required for the liquid to fall from 100 F to 95 F is recorded with a stop watch and this time is a measure of the rate of cooling.

The dry Kata loses its heat by radiation and by convection so for constant velocities the time of cooling is a function of the dry-bulb temperature of the surrounding air. The wet Kata, which has a cloth covering fitted snugly around its bulb, loses heat by radiation, convection, and evaporation, and for constant velocities its rate of cooling is a function of the wet-bulb temperature of the air irrespective of the dry-bulb temperature or relative humidity. It does not follow, however, that the difference in rate of cooling of the dry and the wet Kata is caused by evaporation. A change in the wet-bulb temperature produces a change in the surface temperature of the wet Kata which in turn affects the heat lost by radiation and by convection.

Several precautions should be taken to obtain the best results with this instrument:

1. To obtain velocity readings use the dry Kata since the error in timing is reduced.
2. The instrument should be heated and allowed to cool two or three times before recording the final time of cooling. The first reading is not reliable.
3. All traces of moisture must be removed from the dry Kata before timing to eliminate error introduced by evaporation.
4. Use only the formula applying to a particular instrument. Each Kata receives an individual calibration.

HUMIDITY MEASUREMENT

The sling psychrometer is the recognized standard instrument for determining humidities. In order to obtain accurate readings considerable skill is required on the part of the operator. The wicking and water must be clean and the temperature of the water should be slightly above the wet-bulb temperature of the surrounding air. The psychrometer should be swung rapidly and several and frequent observations should be made to see that the wet-bulb temperature has become stationary before the final reading is noted. Care should be taken that the wet-bulb has reached a minimum temperature, but the wick must still be moist. Standard psychrometric tables should be used[7].

In making wet-bulb measurements below 32 F the same procedure is followed as above 32 F. The water is liquid at the start, but as the sling is operated it will freeze rapidly enough so that in quickly giving up the latent heat of fusion, the indicated wet-bulb temperature may drop below the actual wet-bulb temperature. After the liquid on the bulb has become thoroughly frozen the wet-bulb temperature will rise to normal. A very thin film of ice is more desirable than a thick film. Care must be taken to read the temperatures in the region below 32 F accurately because the spread between the wet- and dry-bulb is small.

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[7]Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew Point; *U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.

In taking humidity readings in ducts it is usually impracticable to use a sling psychrometer. For this work the stationary hygrodeik arranged for bolting on to the side of the duct, with two bulbs extending into the duct, will be found very convenient. Owing to the velocity of the air passing over the bulbs within the duct an accurate reading will be secured, corresponding to that given by the sling psychrometer.

Various forms of humidity recorders are available, some merely recording wet- and dry-bulb temperatures, and others recording relative humidity directly. Any form of wet- and dry-bulb device must have sufficient air velocity over the thermometer bulbs to insure accurate readings; this velocity should be secured by a fan if the air is not itself in motion. A minimum velocity of 900 fpm is usually recommended but velocities from 300 to 1000 fpm have been found suitable under favorable conditions[8]. For extremely low humidities, or for humidity measurements above 212 F, a thermal conductivity method is available[9].

CARBON DIOXIDE DETERMINATION [10]

At ordinary concentrations carbon dioxide is not harmful. The amount of carbon dioxide in the air is a convenient index of the rate of air supply, and of the distribution of the air within rooms. Unequal carbon dioxide concentrations in parts of a room indicate improper air distribution.

The Petterson-Palmquist apparatus has been generally accepted as the standard device for the determination of carbon dioxide in air investigations. The principle involved is the measurement of a given volume of air, the absorption of the contained carbon dioxide in a caustic potash solution, and the remeasurement of the volume of air at the original pressure in a finely graduated capillary tube, the difference in volume representing the absorbed carbon dioxide. (See Report of Committee on Standard Methods for Examination of Air, *American Public Health Association*, Vol. 7, No. 1; *American Journal of Public Health*, Jan., 1917.)

A thermal conductivity method may also be used to measure carbon dioxide in air over a range of 0 to 1.5 per cent[11].

Where field conditions are such that this apparatus may not be conveniently used, as in street cars, air samples may be collected in clean bottles having mercury-sealed rubber stoppers, and these may be subjected to laboratory analysis.

DUST DETERMINATION

Many laboratory methods have been developed to measure the dust in the air. These involve the collection of dust on sticky plates, on filter paper, in water, on porous crucibles, or by electric precipitation, and the subsequent determination of the amount of dust by microscopic counting, weighing, or titration. While there is no standard method, the Hill

[8]Discussion, W. H. Carrier and C. O. Mackey (*A.S.M.E. Transactions*, Vol. 59, No. 6, August, 1937, p. 528-30).

[9]Gas Analysis by Measurement of Thermal Conductivity, by H. A. Daynes (*Cambridge Press*, 1933).

[10]Indices of Air Changes and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[11]Loc. Cit. Note 9.

dust-counter, using a microscope, the impinger[12], using chemical changes in water, and the Lewis sampling tube[13], involving the analytical weighing of a porous crucible, are accepted. All test results should be accompanied by the name of the instrument used as great variation in counts with the different instruments will be obtained. The SOCIETY has developed a code[14] for the testing and rating of air cleaning devices used in general ventilation work.

FLUE GAS ANALYSIS

The analysis of flue gases by chemical means is made with the *Orsat apparatus*. A solution of KOH is used to absorb the CO_2. Free oxygen is absorbed by a mixture of pyrogallic acid and KOH. The solution for absorbing the CO is cuprous chloride. The apparatus consists of a burette surrounded by a water jacket, to receive and measure the volume

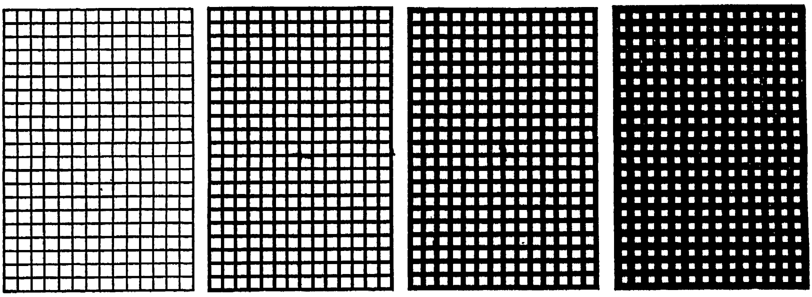

FIG. 1. RINGELMANN SMOKE CHART

of gas. The burette is connected by a manifold of glass to *pipettes* containing liquids for absorbing CO_2, O_2 and CO.

Various forms of automatic indicating and recording gas analysis devices are available, operating on either chemical or physical principles. Such devices are convenient for plant operation.

MEASUREMENT OF SMOKE DENSITY

Relative smoke density is usually measured by comparison with the Ringelmann Chart (Fig. 1). In making observations of the smoke issuing from a chimney, four cards ruled like those in Fig. 1, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point 50 ft from the observer and conveniently in line with the chimney. At this distance, the lines become invisible, and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of

[12]Public Health Bulletin, No. 144, 1925, *U. S. Public Health Service*.

[13]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 270).

[14]A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

the smoke, and makes a record accordingly, noting the time. Observations are made continuously during one minute, and the estimated average density during that minute recorded. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the entire record is plotted on cross-section paper in order to show how the smoke varied in density from time to time.

Smoke recorders are available which give a much more accurate indication of the amount of smoke being produced than does the Ringelmann Chart. They all depend upon projecting a beam of light through the smoke flue or through a separate compartment from which a sample of the flue gas is drawn continuously. The light of the beam which passes through without being absorbed by the smoke is measured to determine the smoke density. Most of these instruments make use of a photoelectric cell or a thermopile to measure the relative amount of light which has not been absorbed. Standard electrical instruments serve for indicating or recording.

MEASUREMENT OF RATE OF HEAT TRANSMISSION

The standard methods of testing built-up wall sections are by means of the *guarded hot-box*[15] and the *guarded hot-plate*[16]. The *Nicholls heat-flow meter* may be used for testing actual walls of buildings.

It would be obviously impossible to determine the air-to-air heat transmission coefficients of every type of wall construction in use with the heat-flow meter, the guarded hot-box or the guarded hot-plate on account of the great amount of time involved. Hence, the method of computing the coefficients from the fundamental constants must be resorted to in most cases. The guarded hot-plate is used to determine the fundamental constants. The heat-flow meter, guarded hot-box and guarded hot-plate tests can be used to good advantage in checking the accuracy of the computed values.

If the hot-box or hot-plate methods are used, tests are usually run under still air conditions, which means there is no wind movement over the surfaces of the wall during the test. In the hot-plate method of test the inside surface coefficient is eliminated by the plates being in direct contact with the wall. In practice, some wind movement over the exterior surface of the wall should always be allowed for; hence, still-air coefficients cannot be used over the outside of the building during the heating season. Moreover, still-air transmission coefficients cannot be corrected to provide for moving-air conditions by applying a single constant factor. Computed coefficients of transmission for various types of construction are given in Chapter 5.

EUPATHEOSCOPE

The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort. See Chapter 41.

[15]Standard Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253) and Report of the Committee on Heat Transmission, *National Research Council.*

[16]Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 65).

PROBLEMS IN PRACTICE

1 ● The hand on a pressure gage attached to a steam line indicates a pressure of 15 lb per square inch and the barometric pressure is 14.7 lb per square inch. What is the absolute pressure, in pounds per square inch, being exerted by the steam?

The absolute pressure exerted by the steam in the pipe is equal to the pressure indicated by the gage plus that exerted by the atmosphere.

Total pressure = 15 + 14.7 = 29.7 lb per square inch.

2 ● What is the corrected barometric pressure of the atmosphere at 32 F when a mercurial barometer reading of 29.51 in. Hg, is determined in a room having a temperature of 91 F?

Substitute in Equation 1. $h = 29.51\ [1 - 0.000101\ (91 - 32)]$.

$h = 29.33$ in. Hg.

3 ● Outline the procedure to be followed in taking room temperatures.

In taking room temperatures, a standard mercury thermometer should be used, with care taken that no part of the observer's body is nearer than 10 in. to the thermometer bulb. The thermometer should be held at least 5 ft away from any window, door or opening; it should be at least 12 in. away from any wall, and should be between 3 and 5 ft from the floor.

4 ● What advantages other than its sensitiveness, has the U tube draft gage or manometer for measurement of low pressures?

Inherent accuracy without calibration and low cost of the essential parts, which are glass tubing and an ordinary scale.

5 ● Are thermocouples as accurate as mercury thermometers?

Within the range which can be measured with both instruments (below 1000 F) either one may be made as sensitive as the service requires. The accuracy of a thermocouple temperature measurement depends chiefly on: (1) an accurate calibration of the wire, (2) the sensitiveness of the electrical instrument, (3) accurate cold-junction control, and (4) proper placement of the sensitive junction.

6 ● When an anemometer is used for measuring the air discharged from a grille or register, does it read the velocity through the gross face area or the velocity through the net free area?

Neither. If either of these velocities is required, it should be calculated by means of Equation 3.

7 ● Do common errors made in humidity determination produce a result that is too high or too low?

A higher relative humidity than the true value is likely to be found, either because there is insufficient velocity over the wet-bulb or because the reading is not taken at the right time.

8 ● What is the purpose of the carbon dioxide determination?

It is an index of the adequacy of fresh air supply and also an indicator of air distribution.

Chapter 45

TERMINOLOGY

Glossary of Physical and Heating, Ventilating and Air Conditioning Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, A.S.H.V.E. Codes

Absolute Humidity: See *Humidity.*

Absolute Pressure: The sum, at any particular time, of the gage pressure and the atmospheric pressure.

Absolute Temperature: The temperature of a substance measured above *absolute zero.*

Absolute Zero: The temperature (−459.6 F) at which the molecular motion of a substance theoretically ceases. This is the temperature at which the substance theoretically contains no heat energy.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second per second.

$$a = \frac{V}{t}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body. In the fps system this is 32.174 ft per second per second.

Adiabatic: An adjective pertaining to or designating variations in volume or pressure not accompanied by gain or loss of heat. When a substance undergoes adiabatic expansion, since it does not receive heat from without, the work which it does is at the expense of its internal energy, and therefore its temperature falls; similarly, when it is adiabatically compressed its temperature rises.

Adsorption: The adhesion of the molecules of gases or dissolved substances to the surfaces of solid bodies, resulting in a concentration of the gas or solution at the place of contact.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Air Infiltration: The inleakage of air through cracks and crevices, and through doors, windows and other openings, caused by wind pressure or temperature difference.

Air Inlet and Air Outlet: Used to designate that point or location in an air handling device or system where air enters or leaves. These terms are misleading and meaningless standing by themselves; must be accompanied by other words to designate location, space, or device which the air is entering or leaving. Thus, *air inlet to a room* may be opening in end of a duct leading from the *air outlet of a fan.*

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure exerted by the atmosphere in all directions, as indicated by a barometer. *Standard atmospheric pressure* is considered to be 14.7 lb per square inch, which is equivalent to 29.92 in. of mercury.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.2 \times 34.5 = 33{,}471.9$ Btu per hour.

British Thermal Unit: The *mean* British thermal unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F. It is substantially equal to the quantity of heat required to raise 1 lb of water from 63 F to 64 F. One Btu $= \frac{1}{3415}$ kwhr.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: The *mean* calorie is $\frac{1}{100}$ of the heat required to raise the temperature of 1 gram of water from *Zero* C to 100 C. It is substantially equal to the quantity of heat required to raise one gram of water from 14.5 C to 15.5 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by

equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distribution ducts. See Chapters 21 and 22.

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Column Radiator: A type of direct radiator. This radiator has not been listed by manufacturers since 1926.

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone *(Average):* The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone (Extreme):* The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 3.)

Concealed Radiator: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, does not *see* the room. Such a device transfers its heat to the room largely by convection air currents.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 F between the two surfaces of the material.

Conductor *(heat):* A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Control: Any manual or automatic device for the regulation of a machine to keep it at normal operation. If automatic, it is considered that the device is motivated by variations in temperature, pressure, time, light, or other influences.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection currents. Such a surface may or may not be enclosed or concealed. When concealed and enclosed

the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator*. See also Chapter 14.)

Corrosive: Having the power to wear away or gradually change the texture or substance of a material.

Decibel: The standard unit for noise or sound intensity. One decibel is equal to ten times the logarithm to the base *e* of the ratio of the sound intensities.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over a 24-hour period, and a temperature of 65 F.

Dehumidify: To remove water vapor from the atmosphere; to remove water vapor or moisture from any material.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = \frac{W}{V}$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Diffuser: A vaned device placed at an air supply opening to direct the air flow.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *Radiator*.

Direct-Return System (*Hot water*)*:* A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)*:* A pipe which carries steam downward to the heating units and into which the condensation from the heating units drain.

Down-Feed System (*Steam*)*:* A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening.

Draft Head (*Top Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe, considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry Air: Air with which no water vapor is mixed. This term is used comparatively, since in nature there is always some water vapor included in air, and such water vapor, being a gas, is dry.

Dry-Bulb Temperature: The temperature of the air indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. See *Wet Return.*

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: The total or impact pressure. This is the sum of the radial pressure and the velocity pressure at the point of measurement.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Enthalpy: Total heat or thermal potential.

Entropy: A ratio, evaluated for practical purposes by dividing the heat content of a unit weight of a substance by its absolute temperature. Useful in examining changes during a heat cycle. Entropy is constant during a reversible adiabatic change of state.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined. It is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = \frac{WV}{gt}$.

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a firepot.

Furnace Volume (*total*): The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as defined above. (A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers.)

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Grille: A perforated covering for an air inlet or outlet usually made of wire screen, pressed steel, cast-iron or plaster. Grilles may be plain or ornamental.

Heat: A form of energy generated by the transformation of some other form of energy, as by combustion, chemical action, or friction. According to the molecular theory, heat consists of the kinetic and potential energy of the molecules of a substance. The addition of heat energy to a body increases the temperature or the kinetic energy of motion of its molecules (*sensible heat*) or increases their potential energy of position but does not increase the temperature, as when melting or boiling occurs (*latent heat*).

Heat Capacity: The amount of heat (Btu or calories) required to raise the temperature of a body of any mass and variety of parts one degree (Fahrenheit or centigrade). This will depend on the masses and specific heats of the various parts of the body.

Therefore

$$S = m_1 s_1 + m_2 s_2 + m_3 s_3 \ldots \text{etc.}$$

where

S is the heat capacity and m_1, m_2, m_3, and s_1, s_2, s_3 stand for the masses and corresponding specific heats of the parts, respectively.

Heating Medium: A substance such as water, steam, air, electricity or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface.*)

Heat of the Liquid: The sensible heat of a mass of liquid above an arbitrary zero.

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft-lb per second or 33,000 ft-lb per minute. (One horsepower = 745.8 watts. In practice this is considered 746 watts.)

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidity: The water vapor mixed with dry air in the atmosphere. *Absolute humidity* refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. *Specific humidity* refers to the weight of water vapor in pounds carried by one pound of dry air. *Relative humidity* is a ratio, usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. Relative humidity is either the ratio of the actual partial pressure of the water vapor in the air to the saturation pressure at the dry-bulb temperature, or the ratio of the actual density of the vapor to the density of saturated vapor at the dry-bulb temperature. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the control of humidity.

Hygrostat: Same as *Humidistat.*

Inch of Water: A measure of pressure which refers to the difference in the heights of the legs of a water-filled manometer.

Insulation *(heat):* A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: See *Heat.*

Laws of Thermodynamics: The *first law* states that the total energy of an isolated system remains constant and cannot be increased or diminished by any physical process whatever. The *second law* states that no change in a system of bodies that takes place of itself can increase the available energy of a system.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second.

$$m = \frac{W}{g}.$$

Mb, Mbh[1]**:** Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The mechanical energy necessary to produce 1 Btu of heat energy. J = 777.5 ft-lb.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol: The unit of weight for gases. It is defined as m lb where m denotes the molecular weight of a gas. For any gas the volume of 1 *mol* at 32 F and standard atmospheric pressure is 358.65 cu ft and the weight of a cubic foot is 0.002788 m lb.

Neutral Zone: The level within a room or building at which the pressure is exactly equal to the outside barometric pressure.

One-Pipe Supply Riser (*steam*)*:* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*hot water*)*:* A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System (*steam*)*:* A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

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[1]*These symbols were approved by the* A.S.H.V.E., June, 1933.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work, expressed in units of horsepower, one of which is equal to 550 ft-lb of work per second, or 33,000 ft-lb per minute.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects "it can see" and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Register: A grille with a built-in multiblade damper or shutter.

Relative Humidity: See *Humidity:* see also discussion of relative humidity in Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*hot water*)*:* A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major subdivision of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to permit egress of air.

Saturated Air: Air containing as much water vapor as it can hold without any condensing out; in saturated air, the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature.

Sensible Heat: See *Heat.*

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The quantity of heat, expressed in Btu, required to raise the temperature of 1 lb of a substance 1 F.

Specific Volume: The volume, expressed in cubic feet, of one pound of a substance. $v = \frac{1}{d} = \frac{V}{W}$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*equivalent*): *Equivalent direct radiation* (EDR). By definition, that amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: As defined by A.S.H.V.E. codes, *standard air* is air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07492 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.921 in. of mercury. The error involved in disregarding the difference between the above two weights is very slight and in most instances may be neglected)

Static Pressure: The compressive pressure existing in a fluid. It is a measure of the potential energy of the fluid.

Steam: Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. Steam in contact with the water from which it has been generated may be *dry saturated* steam or *wet saturated* steam. The latter contains more or less actual water in the form of mist. If steam is heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated.*

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam.*

Supply Mains (*steam*): The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it*, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Synthetic Air Chart: A chart for evaluating the air conditions maintained in a room.

Therm: Symbol used in the gas industry representing 100,000 Btu.

Thermal Resistance: The reciprocal of *conductance.*

Thermal Resistivity: The reciprocal of *conductivity.*

Thermodynamics: The science which treats of the mechanical actions or relations of heat.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The extraction of 12,000 Btu per hour.

Ton Day of Refrigeration: The heat removed by a ton of refrigeration operating for one day; 288,000 Btu.

Total Heat: A thermodynamic quantity, variously called heat content, thermal potential, enthalpy. It is the heat required per unit mass (Btu per pound) to raise a given substance to a given point from an arbitrary datum point. It is the sum of the heat of the liquid, the latent heat, and any miscellaneous heat which may be present.

Total Pressure: The sum of the static and velocity pressures in a fluid. It is a measure of the total energy of the fluid.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*steam or water*)*:* A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*hot water*)*:* A hot water heating system in which the main flow pipe is below the heating units.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit: As applied to heating, ventilating and air conditioning equipment this word means a factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions.

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be *direct*, or *room*, when intended for location, or located in, the treated space; *indirect*, or *remote*, when outside or adjacent to the treated space. They are *ceiling* units when suspended from above,

and *floor* when supported from below. Other descriptive words include *free delivery* when the unit is not intended to be attached to ducts or similar resistance-producing devices, and *pressure* when for use with such ducts. Complete description requires the use of several of these qualifying words or phrases. See Chapter 23.

Up-Feed System (*steam*): A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: The equilibrium pressure exerted by a vapor in contact with its liquid.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The pressure corresponding to the velocity of flow. It is a measure of the kinetic energy of the fluid.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*.)

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: The lowest temperature which a water wetted body will attain when exposed to an air current. This is the temperature of adiabatic saturation.

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. See *Dry Return*.

ABBREVIATIONS [2]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Area	*A*
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiler pressure	bp
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Change in specific volume during vaporization	v_{fg}
Cubic	cu
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[3]	deg or °
Degree centigrade	C
Degree Fahrenheit	F
Degree Kelvin	K
Degree Réaumur	R
Density, Weight per unit volume, Specific weight	*d* or ρ (rho)

$$d = \frac{1}{v}$$

Diameter	*D* or diam
Direct-current (as adjective)	d-c
Distance, linear	*s*
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, Vapor in contact with liquid	*Subscript* g
Entropy (The capital should be used for any weight, and the small letter for unit weight.)	*S* or *s*
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Force, total load	*F*
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal

[2] From compilations of abbreviations approved by the *American Standards Association*, Z, 10 a, c, f, and i. As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word.

[3] It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Head..H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight.)..H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid..h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor..h_g
Heat of vaporization at constant pressure..L or h_{fg}
Horsepower..hp
Horsepower-hour..hp-hr
Inch..in.
Inch-pound..in.-lb
Indicated horsepower..ihp
Indicated horsepower-hour..ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight.)..U or u
Kilogram..kg
Kilowatt..kw
Kilowatthour..kwhr
Length of path of heat flow, thickness..L
Load, total..W
Mass..m
Mechanical efficiency..e_m
Mechanical equivalent of heat..J
Melting point..mp
Meter..m
Micron..μ (mu)
Miles per hour..mph
Minute..min
Molecular weight..mol. wt
Mol..mol
Ounce..oz
Power, Horsepower, Work per unit time..P
Pressure, Absolute pressure, Gage pressure, Force per unit area..p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred..Q
Quality of steam, Pounds of dry steam per pound of mixture..x
Revolutions per minute..rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor..*Subscript* f
Specific gravity..sp gr
Specific heat..sp ht or c
Specific heat at constant pressure..c_p
Specific heat at constant volume..c_v
Specific volume, Volume per unit weight, Volume per unit mass..v
Square foot..sq ft
Square inch..sq in.
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion.)..t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature.)..T or Θ (*capital theta*)
Thermal conductance[4] (heat transferred per unit time per degree)..C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance (heat transferred per unit time per unit area per degree)..C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}.$$

[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples are—conductivity and resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples are—conductance and transmittance. Terms ending *ion* designate rate of heat transfer. Examples are—conduction and transmission.

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* fg
Velocity V
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q
Watt .... w
Watthour .... whr
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

CONVERSION EQUATIONS

Fahrenheit degrees = 9/5 (centigrade degrees) + 32.

Centigrade degrees = 5/9 (Fahrenheit degrees − 32).

Absolute temperature, expressed in Fahrenheit degrees = Fahrenheit degrees + 459.6. In heating and ventilating work, 460 is usually used.

Absolute temperature, expressed in centigrade degrees = centigrade degrees + 273.1.

Power, Heat, and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.33 Btu per pound
1 Btu	= 777.5 ft-lb 0.293 watthours 252.02 mean calories
1 watthour	= 2,655.2 ft-lb 3.415 Btu 3600 joules 860.648 mean calories
1 kilowatthour	= 3,415 Btu 3.52 lb water evaporated from and at 212 F 34.15 lb water raised 100 F
1 mean calorie	= 0.003968 Btu 3.085 ft-lb 0.0011619 watthours
1 kilowatt (1000 watts)	= 1.3405 horsepower 56.92 Btu per minute 44,252.7 ft-lb per minute
1 horsepower	= 0.746 kilowatt 42.44 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,471.9 Btu per hour 9.80 kwhr

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.13368 cu ft
1 British or Imperial gallon	= 277.274 cu in.
1 cu ft	= 7.4805 gal 1728 cu in.
1 cu ft water at 60 F	= 62.37 lb
1 cu ft water at 212 F	= 59.76 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.99 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb
1 long ton	= 2240 lb

Pressure

1 lb per square inch	= 144 lb per square foot 2.0416 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F 1.732 in. water at 62 F

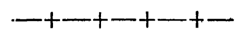
1 atmosphere	= 14.7 lb per square inch 2116.3 lb per square foot 33.974 ft water at 62 F 30 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F	= 0.03609 lb per square inch 0.5774 oz per square inch 5.196 lb per square foot
1 ft water at 62 F	= 0.433 lb per square inch 62.355 lb per square foot
1 in. mercury at 62 F	= 0.491 lb per square inch 7.86 oz per square inch 1.131 ft water at 62 F 13.57 in. water at 62 F

Metric Units

1 cm	= 0.3937 in.
1 in.	= 2.54 cm
1 m	= 3.281 ft
1 ft	= 0.3048 m
1 sq cm	= 0.155 sq in.
1 sq in.	= 6.45 sq cm
1 sq m	= 10.765 sq ft
1 sq ft	= 0.0929 sq m
1 cu cm	= 0.061 cu in.
1 cu in.	= 16.39 cu cm
1 cu m	= 35.32 cu ft
1 cu ft	= 0.0283 cu m
1 liter	= 1000 cu cm = 0.264 gal
1 kg	= 2.2046 lb
1 lb	= 0.4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 980.59 dynes = 0.002205 lb
1 kilometer per hour	= 0.6214 mph
1 gram per square centimeter	= 0.0290 in. mercury, at 0 deg C 0.394 in. water, at 15 C
1 kg per square centimeter (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= 0.03614 lb per cubic inch 62.43 lb per cubic foot
1 dyne	= 0.00007233 poundals
1 joule	= 10,000,000 ergs 0.73767 ft-lb
1 metric horsepower	= 75 kg-m per second 0.986 hp (U. S.)
1 kilogram-calorie (large calorie)	= 1000 gram-calories (small calorie) 3.97 Btu
1 kilogram-calorie per kilogram	= 1.8 Btu per pound
1 gram-calorie per square centimeter	= 3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	= 1.451 Btu per square foot per inch
1 gram-calorie per second per square centimeter for a temperature graduation of 1 deg C per centimeter	= 2903 Btu per hour per square foot for a temperature graduation of 1 deg F per inch of thickness.

SYMBOLS FOR HEATING, VENTILATING AND AIR CONDITIONING DRAWINGS [5]

1. The objects of this standard set of symbols are to insure the correct interpretation of drawings and to conserve drafting room time by establishing simple and unmistakable symbols for the component parts of the heating and ventilating systems. In preparing the list of symbols an effort has been made to follow existing practice in so far as possible but the list cannot be expected to match exactly the existing practice of every drafting room.

1. General Piping
2. Steam Piping
3. Condensate Piping
4. Cold Water Piping
5. Hot Water Piping
6. Air Piping 
7. Vacuum Piping
8. Gas Piping
9. Refrigerant Piping 
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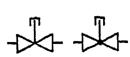
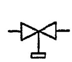
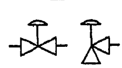
11. Lock and Shield Valve 
12. Reducing Valve 
13. Diaphragm Valve 
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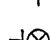
16. Radiator Trap Plan 
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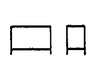
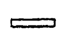
18. Tube Radiator Elevation 
19. Wall Radiator Plan 
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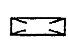
23. Indirect Radiator Plan 
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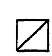
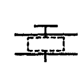
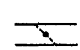
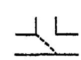
26. Exhaust Duct, Section 
27. Butterfly Damper Plan (or Elevation) 
28. Butterfly Damper Elevation (or Plan) 
29. Deflecting Damper Rectangular Pipe 
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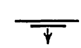
31. Air Supply Outlet 
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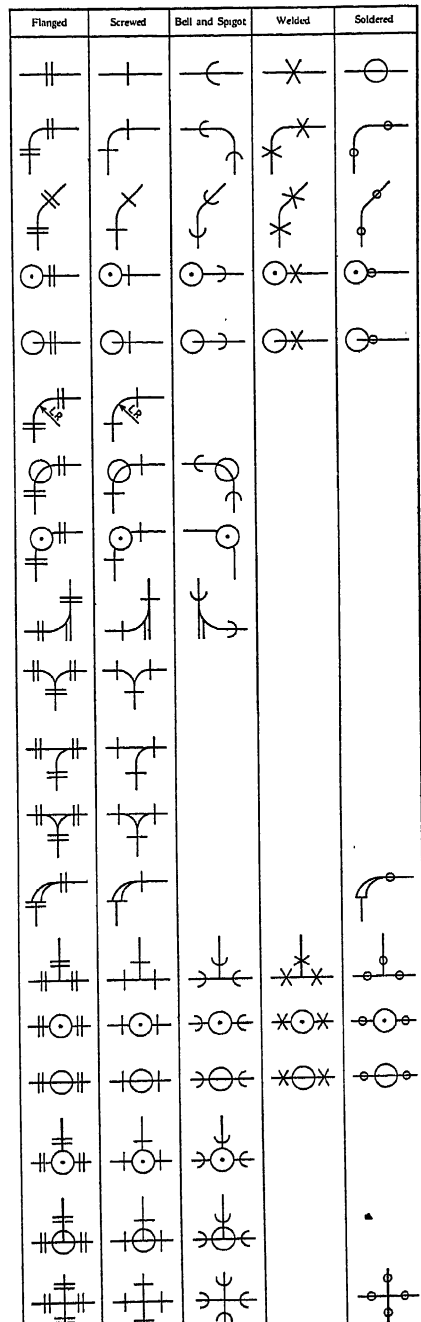
[5]From A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, edition of 1929, and American Standard Drawings and Drafting Room Practice Graphical Symbols (*American Standards Association*, Z14.2—1935).

33. Joint
34. Elbow—90 deg
35. Elbow—45 deg
36. Elbow—Turned Up
37. Elbow—Turned Down
38. Elbow—Long Radius
39. Side Outlet Elbow—Outlet Down
40. Side Outlet Elbow—Outlet Up
41. Base Elbow
42. Double Branch Elbow
43. Single Sweep Tee
44. Double Sweep Tee
45. Reducing Elbow
46. Tee
47. Tee—Outlet Up
48. Tee—Outlet Down
49. Side Outlet Tee—Outlet Up
50. Side Outlet Tee—Outlet Down
51. Cross



52. Eccentric Reducer

53. Reducer

54. Lateral

55. Gate Valve

56. Globe Valve

57. Angle Globe Valve

58. Angle Gate Valve

59. Check Valve

60. Angle Check Valve

61. Stop Cock

62. Safety Valve

63. Quick Opening Valve

64. Float Operating Valve

65. Motor Operated Gate Valve

66. Motor Operated Globe Valve

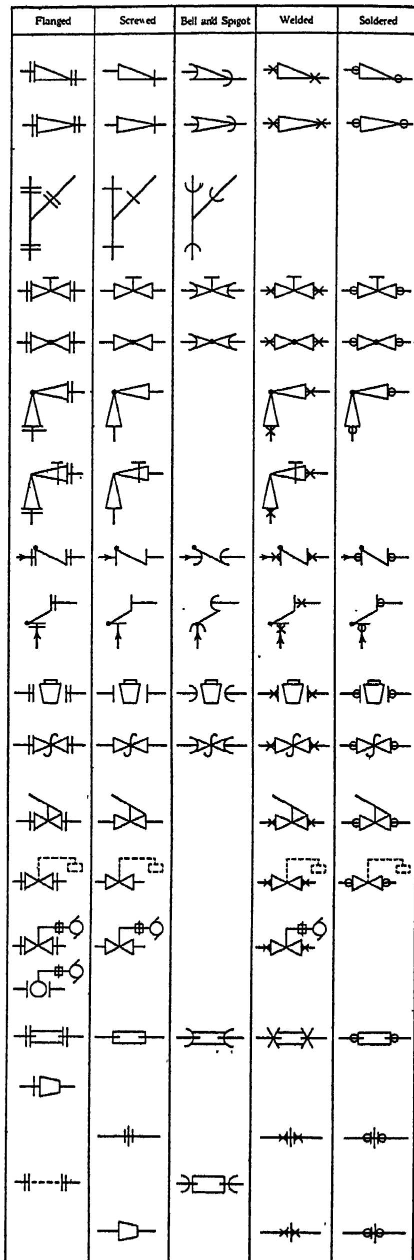
67. Expansion Joint Flanged

68. Reducing Flange

69. Union

70. Sleeve

71. Bushing



A.S.H.V.E. CODES

The following codes and standards relating to the design, installation, testing, rating, and maintenance of materials and equipment used for the heating, ventilation and air conditioning of buildings, have been adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Air Cleaning Devices	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work[a]	June, 1933	A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225
Air Purity	Synthetic Air Chart	June, 1917	A.S.H.V.E. TRANSACTIONS, Vol. 23, p. 607, and THE GUIDE, 1931
Boilers (testing)	Standard and Short-Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)[a]	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 322
Boilers (testing)	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3)[a b]	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 332
Boilers—Oil Fuel (testing)	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[a]	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 23
Boilers (rating)	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[a]	January, 1929 Revised April, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42
Concealed Gravity Type Radiation	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section)[a]	June, 1933	A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237
Convectors	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[a]	January, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Ethics	Code of Ethics for Engineers	January, 1922	A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 6 (See frontispiece THE GUIDE, 1938)
Fans	Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[a]	May, 1923. Revised June, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407[c]
Garages	Code for Heating and Ventilating Garages	June, 1929 Revised January, 1935	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355 A.S.H.V.E. Reprint

[a]Reprints available.
[b]Originally adopted by the *National Boiler and Radiator Manufacturers Association.*
[c]Also, see *Heating, Piping and Air Conditioning,* August, 1931, p. 713.

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Heat Transmission Through Walls	Standard Test Code for Heat Transmission through Walls[a]	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253
Minimum Requirements	Code of Minimum Requirements for Heating and Ventilation of Buildings, Edition–1929	June, 1925	A.S.H.V.E. Codes
Pitot Tube	Code for Use of Pitot Tube	January, 1914	A.S.H.V.E. TRANSACTIONS, Vol. 20, 1914, p. 211
Radiators	Code for Testing Radiators[a]	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters[a d]	January, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators[a]	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25
Vacuum Heating Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps[a]	June, 1934	A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33
Ventilation	Report of Committee on Ventilation Standards[a]	August, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 383

The following Codes and Standards have been endorsed or approved by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	SPONSORED BY	REFERENCE
Air Conditioning Equipment	Standard Method of Rating and Testing Air Conditioning Equipment[e]	*American Society of Refrigerating Engineers*	*American Society of Refrigerating Engineers*, New York, N. Y.
Chimneys	Standard Ordinance for Chimney Construction	*National Board of Fire Underwriters*	Chapter 14, THE GUIDE, 1931
Condensing Units	Standard Method of Rating and Testing Mechanical Condensing Units[e]	*American Society of Refrigerating Engineers*	*American Society of Refrigerating Engineers*, New York, N. Y.
Piping Systems	Identification of Piping Systems[f]	*American Society of Mechanical Engineers*	*Heating, Piping and Air Conditioning*, July, 1929
Warm Air Furnaces	Standard Code Regulating the Installation of Gravity Warm Air Furnaces in Residences	*National Warm Air Heating and Air Conditioning Association*	*National Warm Air Heating and Air Conditioning Association*, Columbus, Ohio

[d]Adopted jointly by the *Industrial Unit Heater Association*, and the A.S.H.V.E.

[e]Proposed code prepared by Joint Committee of *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electric Manufacturers Association* and *Air Conditioning Manufacturers Association*.

[f]Adopted November, 1928, Sponsored by (1) *American Society of Mechanical Engineers*, (2) *National Safety Council*.

CATALOG DATA SECTION

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In this section of The Guide manufacturers of heating, ventilating and air conditioning equipment present their latest developments in apparatus and materials—303 pages of descriptive data, profusely illustrated.

By consistent adherence to a benefit-of-user policy, over a period of 16 years, this Catalog Data Section has become a valuable supplement to the Technical Data Section—a dependable source of information for engineers, architects, contractors, and others in this field of industry.

Products are grouped in alphabetical arrangement so that a specific type of equipment or material may be located readily by reference to the page headings—Boilers, Heaters, Insulation, etc.—on pages 843-848. On pages 1165-1188 is a complete Index to Modern Equipment.

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Hoffman Specialty Co., Inc., 1112-1113
Ingersoll-Rand Company, 862-863
Nash Engineering Co., 1086-1087

PUMPS, Vacuum

American Marsh Pumps, Inc., 1082
Chicago Pump Co., 1084
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Hoffman Specialty Co., Inc., 1112-1113
Ingersoll-Rand Company, 862-863
Nash Engineering Co., 1086-1087
Sterling Engineering Co., 1121

PYROMETERS, Portable and Stationary

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Illinois Testing Laboratories, Inc., 1014
Leeds & Northrup Company, 1015
Minneapolis-Honeywell Regulator Co., 1148-1149
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Aerofin Corporation, 996-998
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RADIATION, Brass

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G & O Manufacturing Co., 999
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RADIATION, Cast-Iron

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Unit Heater and Cooler Co., 994
United States Radiator Corporation, 950-951
Utica Radiator Corp., 920-921
Weil-McLain Company, 952

RADIATION, Copper

Aerofin Corporation, 996-998
American Radiator Company, 884-885, 940-943, 1061
C. A. Dunham Co., 1108-1109

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

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Roll of Membership

AMERICAN SOCIETY *of*
HEATING *and* VENTILATING ENGINEERS

1938

Contains Lists of Members Arranged Alphabetically and Geographically, also Lists of Officers and Committees, Past Officers and Local Chapter Officers

Corrected to January 1, 1938
Published at the Headquarters of the Society
51 Madison Avenue, New York, N. Y.

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AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

51 Madison Ave., New York, N. Y.

1937-38

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University of Oklahoma, Norman, Okla.

Secretary, E. W. GRAY
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Secretary, H. R. ROTH
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Secretary, M. N. MUSGRAVE
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Secretary, C. B. EASTMAN
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Brookline, Upper Darby, Pa.

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P. O. Box 1046

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Carnegie Inst. Tech.

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Secretary, D. J. FAGIN
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1234 South Grand

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Secretary, W. H. BADGETT
Texas Engrg. Experiment Station,
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Roll of Membership

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1938

(Corrected to January 1, 1938)

HONORARY MEMBERS

BALDWIN, WM. J. (1915), New York, N. Y. (Deceased May 7, 1924.)
BILLINGS, DR. J. S. (1896), New York, N. Y. (Deceased March 10, 1913.)
BOLTON, REGINALD PELHAM (1897), New York, N. Y.
BRECKENRIDGE, L. P. (1920), North Ferrisburg, Vt.
GORMLY, JOHN (Charter Member), Norristown, Pa. (Deceased January 31, 1929.)
NEWTON, C. W. (Charter Member), Baltimore, Md. (Deceased August 6, 1920.)
HOOD, O. P. (1929), Washington, D. C. (Deceased April 22, 1937.)
JELLETT, STEWART A. (Charter Member), (Presidential Member), Philadelphia, Pa. (Deceased April 5, 1935.)

LIST OF MEMBERS Arranged Alphabetically

(Asterisk indicates authorship of papers)

(*M* 1923; *A* 1918; *J* 1916) indicates, Election as Member 1923; Associate 1918; Junior 1916.
(Pres. 1923) indicates, Elected President in 1923 and is now a Presidential Member.

A

ABRAMS, Abraham (*M* 1927; *J* 1924), Pres., Abbey Heating Co., Inc., 81 Centre Ave., and (for mail), 100 Clove Rd., New Rochelle, N. Y.

ACHESON, Albert R. (*M* 1919), Consulting Engr. (for mail), 501 Fekel Theatre Bldg., and 852 Ostrom Ave., Syracuse, N. Y.

ADAMS, Benjamin (*M* 1919), Commercial Mgr. (for mail), American Blower Corp., Room 781 Broad Street Station Bldg., and 3006 W. Coulter St., Queen Lane Manor, Philadelphia, Pa.

ADAMS, Benjamin C., Jr. (*S* 1936), Engrg. Student (for mail), 724 Chautauqua, Norman, Okla., and 5127 Sunset Drive, Kansas City, Mo.

ADAMS, Bruce P. (*A* 1930), Gen. Mgr. (for mail), McDonnell & Miller, 400 N. Michigan Ave., and 1432 Rancher Ave., Chicago, Ill.

ADAMS, Charles W. (*M* 1920), Salesman, U. S. Radiator Corp., 1221 West 11th St., Kansas City, Mo.

ADAMS, Harold E. (*M* 1930), Chief Engr. (for mail), Nash Engineering Co., Wilson Rd., South Norwalk and Merrill Heights, Norwalk, Conn.

ADAMS, Neil D. (*M* 1929; *A* 1925; *J* 1922), Supt., Franklin Heating Station (for mail), 220 Second Ave., S.W., and 836 Eighth Ave., S.W., Rochester, Minn.

ADDAMS, Homer (*Charter Member; Life Member*), (*Presidential Member*), (Pres., 1924; 1st Vice-Pres., 1923; Treas., 1915-1922; Council, 1915-1925), Pres., Kewanee Boiler Co., Inc., and Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, N. Y.

ADLAM, T. Napier (*M* 1932), Vice-Pres. and Gen. Mgr., Sarco Mfg. Co., 183 Madison Ave., New York, N. Y., and (for mail), 64 Wellington Ave., West Orange, N. J.

ADLER, Alphonse A.* (*M* 1921), Consulting Engr., 35 Stewart Ave., Arlington, N. J.

ADLER, Jack C. (*A* 1937; *J* 1936), Sales Mgr., Air Cond. Dept., Frigidaire Corp., 224 West 57th St., New York, and (for mail), c/o B. W. Adler, 6952 Groton St., Forest Hills, L. I., N. Y.

ADSHEAD, Bernard (*J* 1936), Tech. Director, National Air Conditioning & Humidifying Co., Ltd., 46 Britannic Bldg., Manchester, and (for mail), 53 Shamrock Rd., Birkenhead, Cheshire, England.

AEBERLY, John J.* (*M* 1928), (Council, 1937), Chief of Div. of Htg., Vtg. and Ind. Sanitation, Chicago Board of Health, 707 City Hall, and (for mail), 6225 N. Newcastle Ave., Norwood Park P. O., Chicago, Ill.

AHEARN, William J. (*M* 1929), Htg. and Vtg. Engr., 21 Lake Rd., Cochituate, Mass.

AHLBERG, Henry B. (*A* 1938; *J* 1933), Chief Engr., Anderson Products Co., 17 Tudor St., Cambridge, and (for mail), 146 Orlando St., Mattapan, Mass.

AHLFF, Albert A. (*M* 1923; *A* 1918), Branch Mgr. (for mail), National Radiator Corp., 2124 Arch St., Philadelphia, and 43 Rock Glen Rd., Overbrook Hills, Philadelphia, Pa.

AIKMAN, Joseph M. (*M* 1936), Consulting Air Cond. Engr., 2351 N. Cleveland Ave., Chicago, Ill.

AITKEN, James (*A* 1935), 740 Gladstone Ave., Windsor, Ont., Canada.

AKERMAN, Joseph Reid (*J* 1937), Htg. and Air Cond. Engr. (for mail), Phoenix Oil Co., 700 Twiggs St., and 831-15th St., Augusta, Ga.

AKERS, George W. (*M* 1929), Secy.-Treas., George W. Akers Co., 16525 Woodward Ave., Detroit, and (for mail), R. F. D. No. 2, Birmingham, Mich.

ALBRECHT, Henry P. (*J* 1937), Engr. (for mail), Reinhard Bros. Co., Inc., 11 S. Ninth St., and 3521 Park Ave., Minneapolis, Minn.

ALEXANDER, Samuel W. (*M* 1935), Mgr. Htg. Div., James Morrison Brass Co., 276 King St., S.W., and (for mail), 124 Kingsmount Park Rd., Toronto, Ont., Canada.

ALFAGEME, Braulio (*M* 1935), Engr., Mgr., B. Alfageme, Almagro 1, Madrid, Spain.

ALFSEN, Nikolai (*M* 1933), Civil Engr., Alfsen & Gunderson, A/S Oslo, Prinsensgate 2b, and (for mail), Shabekk near Oslo, Norway.

ALGREN, Axel B.* (*M* 1930), Asst. Prof. Mech. Engr., University of Minnesota, Exp. Engrg. Lab., and (for mail), 5109–17th Ave., S., Minneapolis, Minn.

ALLAN, William (*A* 1937), Pres. and Treas. (for mail), Allan Engineering Co., 724 E. Mason St., and 2735 N. Farwell Ave., Milwaukee, Wis.

ALLAIRE, Lucien (*J* 1937), Drainage Engr., Department of Agriculture, Quebec City, and (for mail), 2182 Sherbrooke St., E., Montreal, Canada.

ALLCUT, Edgar A.* (*M* 1937), Prof. of Mech. Engrg. (for mail), University of Toronto, Dept. of Mech. Engrg., and 48 Foxbar Rd., Toronto, Ont., Canada.

ALLEN, A. Walter (*M* 1936), Sales Engr., Pease Foundry Co., Ltd., Toronto, and (for mail), 151 Glen Ave., Ottawa, Ont., Canada.

ALLEN, Carl V. (*M* 1937), Engrg. Mgr., Midwest Air Conditioning Corp., 1909 Washington, and (for mail), 5562 Clemens, St. Louis, Mo.

ALLEN, DeWitt M. (*M* 1936; *J* 1922), Dist. Mgr. (for mail), Ilg Electric Ventilating Co., 310 Board of Trade Bldg., and 5700 Olive St., Kansas City, Mo.

ALLEN, William W. (*A* 1936), Pres. (for mail), American Coolair Corp., Box 2300, Jacksonville, and DaVinci St., Venetia, Fla.

ALLONIER, Howard R. (*A* 1936), Dist. Mgr. (for mail), Buckeye Blower Co., Box 195 (425 W. Town St.), Columbus, and R. F. D. No. 1, Powell, Ohio.

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DICKSON, Robert B. (*M* 1919), Pres. (for mail), Kewanee Boiler Corp., Franklin St., and Q Tracks, and 145 E. Division St., Kewanee, Ill.
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HOBBS, J. Clarence (*M* 1920), Gen. Mgr., and Mfg. Operations (for mail), Diamond Alkali Co., and 60 Wood St., Painesville, Ohio.

HOBBS, William S. (*A* 1936), Owner and Mgr. (for mail), P. O. Box 269, and 327 Park Ave., Swarthmore, Pa.

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HUGONIOT, Victor E. (*M* 1935), Zone Engrg. Mgr. (for mail), Airtemp Sales Corp., 3717 Washington Ave., and 1347 Kingsland Ave., St. Louis, Mo.
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Ward, H. H.

Miami Beach—
Friedman, D. H., Jr.

Orlando—
Lyle, E. T.
Porter, C. W.

West Palm Beach—
Hodeaux, W. L.

GEORGIA

Atlanta—
Baker, C. T.
Barnes, L. L.
Beechler, J. S.
Boyd, S. W.
Brockinton, C. E.
Clare, F. W.
Cole, C. B.
Driscoll, M. G.
Foss, E. R.
Gouedy, K. E.
Gunnell, G. T.
Hahn, R. F.
Kagey, I. B., Jr.
Kelley, R. D.
Kent, L. F.
Klein, E. W.
Krayenhof, H. G.
McCain, H. K.
McKinney, W. J.
Pounds, C. A.
Sudderth, L., Jr.
Templin, C. L.
Tucker, T. T.

Augusta—
Akerman, J. R.
Arndt, H. W.

Decatur—
Sherman, W. P.

Savannah—
Hamlin, J. B., Jr.

HAWAII

Honolulu—
Manning, C. E.
Petersen, S. E.

ILLINOIS

Alton—
Carlock, M. F.

Bloomington—
MaGirl, W. J.
Nesmith, O. E.
Scholl, H. O.
Soper, H. A.
Whitmer, R. P.

Champaign—
Hintz, H. P.
Ransom, C. F.
Strakosh, W. C.
Thetford, J. E.

Chicago—
Adams, B. P.
Aeberly, J. J.
Aikman, J. M.
Ammerman, A. S., Jr.
Arenberg, M. K.
Baker, W. H., Jr.
Bamond, M. J.
Baumgardner, C. M.
Baur, J. W.
Becker, W. A.
Beery, C. E.
Benson, B. C.
Bernstrom, B.
Bishop, M. W.
Black, F. C.
Boehmer, A. P.
Bolte, E. E.
Borling, J. R.
Bowles, E. N.
Boyle, J. R.
Bracken, J. H.
Braun, L. T.
Brigham, C. M.
Brightly, F. C., Jr.
Brocha, J. F.
Brooke, I. E.
Broom, B. A.
Brown, A. P.
Brown, T.
Burnam, C. M., Jr.
Casey, B. L.
Chapin, H. G.
Christman, W. F.
Christopherson, A. E.
Clegg, R. R.
Cochran, C. C.
Crone, C. E., Jr.
Crump, A. L.
Cunningham, T. M.
Cutler, J. A.
Dasing, E.
Dauber, O. W.
DeLand, C. W.
Dolson, C. N.
Dunham, C. A.
Emmert, L. D.
Ericsson, E. B.
Eskin, S. G.
Fatz, J. L.
Finan, J. J., Sr.
Fleming, J. P.
Fleming, T. F.
Frank, J. M.
Gardner, W., Jr.
Gaylord, F. H.
Getschow, R. M.
Gibbs, F. C.
Goelz, A. H.
Gossett, E. J.
Gothard, W. W.
Gotschall, H. C.
Graham, E. W.
Graves, W. B.
Haas, S. L.
Haines, J. J.
Hale, J. F.
Hanley, T. F., Jr.
Hart, H. M.
Hattis, R. E.
Hayden, C. F.
Hayes, J. J.
Hebley, H. F. J.
Heckel, E. P.
Herlihy, J. J.
Hess, D. K.
Hill, E. V.
Hinckley, H. B.
Hines, J. C.
Horner, S. D.
Howard, F. L.
Howatt, J.
Howell, L.
Hubbard, G. W.
Hustoel, A. M.
Isett, W. M.
Jenson, J. S.
Johns, H. B.
Johnson, C. W.
Keating, A. J.
Keeney, F. P.
Kehm, H. S.
King, A. C.
Knudsen, W. R.
Krez, L.
Kyle, W. J.
Lagodzinski, H. J.
La Roi, G. H., II
Larson, C. P.
Lauterbach, H., Jr.
Leuthesser, F. W., Jr.
Lewis, S. R.
Lindsay, G. W., Jr.
Linn, H. R.
Lockhart, H. A.
Luders, R. H.
Mabley, L. C.
Machen, J. T.
Malone, D. G.
Malvin, R. C.
Manny, J. H.
Marschall, P. J.
Martin, A. B.
Matchett, J. C.
Mathis, E.
Mathis, H.
Mathis, J. W.
May, E. M.
May, M. F.
McCauley, J. H.
McClellan, J. E.
McDonnell, E. N.
McDonnell, J. E.
Medow, J.
Mertz, W. A.
Miller, F. A.
Miller, R. T.
Milliken, J. H.
Mueller, H. C.
Murphy, E. T.
Murphy, W. A.
Narowetz, L. L., Jr.
Neiler, S. G.
Newport, C. F.
Nightingale, G. F.
Offen, B.
Olsen, C. F.
Olson, B.
Paul, L. O.
Peller, L.
Pickett, C. A.
Pitcher, L. J.
Pope, S. A.
Powers, F. W.
Prentice, O. J.
Price, C. E.
Priester, G. B.
Rasmussen, R. P.
Raymond, F. I.
Reger, H. P.
Reid, H. P.
Rietz, E. W.
Rottmayer, S. I.
Runkel, C.
Russell, E. A.
Ryerson, H. E.
Scheidecker, D. B.
Schuetz, C. C.
Schuler, W. B.
Schwartz, H. J.
Schweim, H. J.
Seelig, L.
Shilling, H. C.
Shultz, E.
Solstad, L. L.
Sommerfield, S. S.
Spielman, G. P.
Stacy, L. D.
Stannard, J. M.
Stermer, C. J.
Stokes, A.
Sutcliffe, A. G.
Swanson, N. W.
Thinn, C. A.
Thomas, R. H.
Thommen, A. A.
Tobin, J. F.
Tornquist, E. L.
Trumbo, S. M.
Turner, G. G.
Van Alsburg, J. H.
Vernon, J. R.
Walters, W. T.
Washington, L. W.
Weil, M.
Weil, M. I.
Weinfeld, C.
Weinshank, T.
Wendt, E. H.
White, E. B.
Whittington, J. A.
Widdowfield, A. S.
Wilson, W. H.
Wolff, P. P.
Zack, H. J.
Zimmerman, A. H.

Chicago Heights—
Boyar, S. L.

Cicero—
Brown, N. A.

Decatur—
Shorb, W. A.

East St. Louis—
Cover, E. B.

Edwardsville—
Blackmore, J. J.

Elmhurst—
Jones, D. J.

Evanston—
Horton, H. F.
Maccubbin, H. A.
Miller, J. E.

Glencoe—
Hornung, J. C.

Glen Ellyn—
Parsons, L. D., Jr.
Sherman, V. L.

Homewood—
Wilkinson, F. J.

Hoopeston—
Moore, D. R.

Kewanee—
Bronson, C. E.
Dickson, R. B.
Hartman, J. M.
Pursell, H. E.

LaGrange—
Eaton, B. K.

Lake Forest—
Scudder, B.

Moline—
Beling, E. H.
Johnson, W. G.
Nelson, H. W.
Nelson, R. H.
Otis, G. E.
Sharp, H. C.

Mt. Vernon—
Benoist, L. L.
Benoist, R. E.

Oak Park—
Blanding, G. H.
Fitzgerald, M. J.
Smith, R. H.
Uhlhorn, W. J.

Park Ridge—
Bevington, C. H.
Kuechenberg, W. A.
Moore, R. E.
Spielmann, H. J.

Peoria—
Baird, S. A.
Farnsworth, J. G.
Fox, E. L.
Hauer, F.
Logan, T. M.
Meyer, F. L.
Weeks, P.

Peru—
Gallagher, P.

Rockford—
Braatz, C. J.
Dewey, R. P.
Drake, G. F.
Merwin, G. E.
Pruden, B.
Stewart, D. J.

Urbana—
Bowditch, R. P.
Broderick, E. L.
Compton, W. E.
Fahnestock, M. K.
Konzo, S.
Kratz, A. P.
Severns, W. H.
Willard, A. C.
Wright, W. J.

Villa Park—
Armspach, O. W.

Waukegan—
Killian, T. J.

Western Springs—
Washington, G.

Winnetka—
Close, P. D.
Killian, V. J.
Mittendorff, E. M.

Zion—
Baughman, L. R.

INDIANA

Evansville—
Bulleit, C. R.
Grossman, F. A.

Goshen—
Shaw, B. E.

Huntington—
Redrup, W. D.
Smith, G. W.

Indianapolis—
Ammerman, C. R.
Fenstermaker, S. E.
Fillo, F. B.
Hagedon, C. H.
Hamerski, F. D.
Hayes, J. G.
Hildreth, E. S.
Mulloy, E.
Niesse, J. H.
Poehner, R. E.
Supple, G. B.
Voorhees, G. A.

Lafayette—
Voigt, R. N.

LaPorte—
Shrock, J. H.
Truitt, G. S.

Lawrenceburg—
Bechtol, J. J.

Michigan City—
Stockwell, W. R.
Westphal, N. E.

Muncie—
Pfriem, P. G.
Price, C. F.
Waggoner, J. H.

Peru—
Thrush, H. A.

St-Mary-of-the-Woods—
Bisch, B. J.

Vincennes—
Wissing, C. B.

Wabash—
Shivers, P. F.

West Lafayette—
Hoffman, J. D.

IOWA

Ackley—
Nelson, G. O.

Ames—
Norman, R. A.

Cedar Rapids—
Friedline, J. M.

Des Moines—
Daubert, L. L.
Elbert, B. F.
Stiles, G. S.
Vaughn, F. R.
Walters, A. L.

Sioux City—
Hagan, W. V.

Waterloo—
Todd, M. L.
Winterbottom, R. F.

KANSAS

Fort Leavenworth—
Foley, D. F.
Gamble, C. L.

Hutchinson—
Mann, A. R.
Stevens, H. L.

Lawrence—
Machin, D. W.
Sluss, A. H.

Neodesha—
Berzelius, C. E.

Russell—
Danielson, E. B.

Salina—
Ryan, W. F.

Wichita—
Leverance, H. J.
Olson, G. E.

KENTUCKY

Anchorage—
Brooks, H. B.

Cold Spring—
Ward, F. J.

Fort Thomas—
Leupold, G. L.
Reik, R. C.
Stevens, W. R.

Lexington—
Bozeman, R. W.
Cabot, M. A.
May, J. W.
O'Bannon, L. S.
Wallace, W. M., II
West, P.

Louisville—
Danielson, W. A.
Fitch, H. M.
Grabensteder, L.
Graham, J. M.
Groot, H. W.
Hellstrom, J.
Murphy, H. C.
White, T. G., Jr.

LOUISIANA

New Orleans—
Blum, H., Jr.
Gamble, C. B.
Gammill, O. E., Jr.
Herman, N. B.
Kaufman, C. W.
May, G. E.
McLaren, F. S.
Moses, W. B., Jr.
Perkins, R. C.
Ryan, J. D.
Salzer, A. R., Jr.
Seidel, G. E.

Shreveport—
Fitzgerald, W. E.
Zwally, A. L.

MAINE

Bangor—
Prince, R. F.

Lewiston—
Fowles, H. H.

Portland—
Fels, A. B.
Merrill, C. J.
Mitchell, C. H.

MARYLAND

Annapolis—
Gale, H. A.

Baltimore—
Collier, W. I.
Crosby, E. L.
Dorsey, F. C.
Dull, E. J.
Hunt, M.
Jex, J., Jr.
Leilich, R. L.
McCormack, D.
Page, V. C.
Posey, J.
Powers, E. C.
Seiter, J. E.
Shepard, J. deB.
Sklarevski, R.
Smoot, T. H.
Steele, M. G.
Stehl, H. V.
Vance, L. G.
Viessman, W.
Vincent, P. J.
Whiteley, S. M.

Bethesda—
Goodwin, E. W.
Stock, C. S.
Terhune, R. D.
Vaughan, J. G., Jr.

Brooklyn Park—
Rodgers, J. S.

College Park—
Gifford, W. R.

Ellicott City—
Tibbets, J. C.

Rockville—
Brunett, A. L.

Silver Spring—
Black, F. M.
Stack, A. E.

MASSACHUSETTS

Arlington—
Shaw, N. J. H.

Arlington Heights—
Tarr, H. M.

Auburndale—
Frederick, H. W.

Belmont—
Spence, R. A.

Boston—
Archer, D. M.
Barron, J. T.
Berchtold, E. W.
Boyden, D. S.
Brinton, J. W.
Brissette, L. A.
Bryant, A. G.
Bullock, T. A.
Cummings, C. H.
Dickson, G. P.
Donohoe, J. B.
Drinker, P.
Edwards, D. J.
Foulds, P. A. L.
Franklin, R. S.
Gleason, G. H.
Hashagen, J. B.
Jennings, R. A.
Jennings, W. G.
Keefe, E. T.
Kelley, J. J.
Kellogg, A.
Keyes, R. E.
Kimball, C. W.
Little, D. H.
Macrow, L.
McCoy, T. F.
Merrill, F. A.
Millard, J. W.
Miller, J. F. G.
Mullen, T. J., Jr.
Nee, R. M.
Osborne, M. M.
Palmer, R. T.
Plunkett, J. H.
Rydell, C. A.
Spoerr, F. F.
Stetson, L. R.
Swaney, C. R.
Tuttle, J. F.
Waterman, J. H.
Whitten, H. E.
Yaglou, C. P.

Bridgewater—
Beaulieu, A. A.

Cambridge—
Daitsh, A.
Flint, C. T.
Gerrish, G. B.
Haddock, I. T.
Holt, J.
Hoyt, C. W.
Longwell, J. S.
MacDonald, E. A.
Moore, H. C.
Peterson, C. M. F.
Sheffield, R. A.
Turner, J.
Wilkes, G. B.

Cochituate—
Ahearn, W. J.

Dalton—
Dakin, H. W.

Dorchester—
Brayman, A. I.
Goodrich, C. F.
Hosterman, C. O.
Shaer, I. E.
Voye, V. J.

East Milton—
Austin, W. H.

Fall River—
Fenner, E. M.

Fitchburg—
Earley, T. J.
Illig, W. R.
Karlson, A. F.
McKittrick, P. A.

Foxboro—
Meakin, J. B.

Harwich Port—
Maxwell, G. W.

Holyoke—
Colby, C. W.

Hyde Park—
Bartlett, A. C.
Ellis, F. R.
Wolf, J. C.

Lawrence—
Bride, W. T.

Leominster—
Kern, R. T.

Lynn—
Farrow, H. L.
Feehan, J. B.

Mattapan—
Ahlberg, H. B.

Milton—
Corey, G. R.

Needham—
Park, C. D.
Webb, J. S.

Newton Centre—
Murray, J. J.

Newtonville—
Emerson, R. R.
Jones, W. T.
McMurrer, L. J.

Pittsfield—
Wagner, E. A.

Quincy—
Stone, E. R.

Reading—
Ingalls, F. D. B.

Roslindale—
Larson, C. W.

Roxbury—
Madden, J. J.

Sharon—
Nelson, A. W.

South Hamilton—
Mandell, T. P.

South Boston—
Hilliard, C. E.

Springfield—
Blair, H. A.
Cross, R. E.
Holmes, R. E.
Leland, W. B.
Murphy, W. W.

Swampscott—
Knowles, M. G.
Oates, W. A.

Watertown—
Wiegner, H. B.

Wellesley Hills—
Barnes, W. E.

West Roxbury—
Christie, A. Y.
McCafferty, J. E.
McPherson, W. A.

Woburn—
Parker, P.

Wollaston—
Hodgdon, H. A.

Worcester—
Wechsberg, O.

MICHIGAN

Ann Arbor—
Backus, T. H. L.
Bichowsky, F. R.
Emswiler, J. E.
Marin, A.

Battle Creek—
Christenson, H.

Birmingham—
Hadjisky, J. N.
Hyde, E. F.
Root, E. B.

Dearborn—
King, H. K.

Detroit—
Akers, G. W.
Annas, H. C.
Arnoldy, W. F.
Baker, R. H.
Baldwin, W. H.
Barth, H. E.
Barton, J.
Bishop, F. R.
Blackmore, F. H.
Boales, W. G.
Brown, R. E. G.
Brown, W.
Burks, R. H.
Clark, E. H.
Connell, R. F.
Coon, T. E.
Cummins, G. H.
Darlington, A. P.
Dauch, E. O.
Deppmann, R. L.
Dickenson, F. R.
Dubry, E.
Eckert, E. K.
Estep, L. G.
Falk, D. S.
Feely, F. J.
Feinberg, E.
Flink, C. H.
Ford, E. F.
Giguere, G. H.
Halleck, L. P.
Hamaker, A. C.
Harrigan, E. M.
Hesselschwerdt, A. L., Jr.
Heydon, C. G.
Hogan, E. L.
Hubbard, N. D.
Hughson, H. H.
Hutzel, H. F.
Kaufman, H. J.
Keyser, H. M.
Kilner, J. S.
Kincaide, M. C.
Kirkpatrick, A. H.
Knapp, A. E.
Knibb, A. E.
Lee, J. A.
Lewis, G. M.
Linsenmeyer, F. J.
Long, D. J.
Luty, D. J.
Maier, G. M.
Martel, C. L.
Marzolf, F. X.
McConachie, L. L.
McCrea, J. B.
McGaughey, H. M.
McGeorge, R. H.
McIntire, J. F.
McLean, D.
Metcalfe, C.
Meyer, J., Jr.
Miller, R. E.
Milward, R. K.
Morse, C. T.
Morse, L. S., Jr.
O'Gorman, J. S.
Old, W. H.
Paetz, H. E.
Parrott, L. G.
Partlan, J. W.
Patterson, F. H.
Pavey, C. A.
Pike, W.
Purcell, F. C.
Randall, R. D.
Randall, W. C.
Russell, W. A.
Sanford, S. S.
Schechter, J. P.
Schmidt, K., Jr.
Shea, M. B.
Sheley, E. D.
Smith, W. O.
Snell, E.
Snyder, J. W.
Soeters, M.
Spitzley, R. L.
Spurgeon, J. H.
Sterner, D. S.
Stites, R., Jr.
Taylor, H. J.
Toonder, C. L.
Tuttle, G. H.
Van Nouhuys, H. C.
Volberding, L. A.
Wald, G. H.
Walker, J. H.
Weinert, F. C.
Whelan, W. J.
White, E. S.
Whitt, S. A.
Wigle, B. M.
Winans, G. D.

Dowagiac—
Cunningham, J. S.
Torr, T. W.

East Lansing—
Miller, L. G.
Pesterfield, C. H.

Grand Rapids—
Bradfield, W. W.
Bratt, H. D.
Epple, A. B.
Graff, W. F.
Marshall, O. D.
Morton, C. H.
Osberger, T. L.
Stafford, T. D.
Todd, S. W.
Warren, F. C.
Worthing, S. L.
Ziesse, K. L.

Highland Park—
Harrower, W. C.

Houghton—
Seeber, R. R.

Iron Mountain—
Eisele, L. G.

Jackson—
Gerhard, D. H.

Kalamazoo—
Brinker, H. A.
Downs, S. H.
McConner, C. R.
Metzger, H. J.
Schlichting, W. G.
Temple, W. J.
Wilson, R. W.

Lansing—
Chrouch, R. B.
Haan, R. B.
McLouth, B. F.
Parsons, R. A.
Vanderlip, P. J.
Witmer, H. S.

Mt. Clemens—
Bailey, E. P., Jr.

Muskegon—
Young, H. J.

Muskegon Heights—
Reid, H. F.

Pontiac—
Singleton, J. H.

Port Huron—
Blessed, W. A.

Royal Oak—
Burch, L. A.
Helmrich, G. B.

Saginaw—
Witheridge, D. E.

MINNESOTA

Bayport—
Swanson, E. C.

Duluth—
Foster, C.

Hibbing—
Miller, L. L.

Minneapolis—
Albrecht, H. P.
Algren, A. B.
Anderson, S. H.
Armstrong, R. W.
Bell, E. F.
Bensen, C. L.
Betts, H. M.
Bjerken, M. H.
Bredesen, B. P.
Burns, E. J.
Burritt, C. G.
Campbell, R. L.
Carlson, C. O.
Comb, F. R., Jr.
Cooper, T. E.
Copperud, E. R.
Cumming, F. J.
Dahlstrom, G. A.
Davidson, J. C.
Dovolis, N. J.
Doxey, H. E.
Edelman, B. P.
Fergestad, M. L.
Forfar, D. M.
Francis, P. E.
Gausewitz, W. H.
Gerrish, H. E.
Gordon, E. B., Jr.
Gross, L. C.
Hall, J. R.
Hanson, L. P.
Harris, J. B.
Hawkinson, C. F.
Helstrom, H. G.
Hitchcock, P. C.
Huch, A. J.
Johnson, L. H.
King, R. L.
Kingsland, G. D.
Knapp, D. S.
Knowles, E. L.
Kuehn, W. C.
Lange, F. F.
Legler, F. W.
Lilja, O. L.
Lund, C. E.
Miller, L. B.
Mills, H. C.
Morgan, G. C.
Morton, H. S.
Ogard, N. L.
Orr, G. M.
Petersen, C. P.
Poucher, R. C.
Roberts, H. P.
Roberts, J. R.
Rowley, F. B.
Schad, C. A.
Schernbeck, F. H.
Schultz, A. W.
Schwantes, A. R.
Seelert, E. H.
Spencer, J. B.
Stiller, F. W.
Sturm, W.
Sundell, S. S.
Sutherland, D. L.
Swenson, J. E.
Uhl, E. J.
Uhl, W. F.
Wallace, H. P., Jr.
Welter, M. A.
Willis, L. L.

Owatonna—
Anderson, G. A. M.

Rochester—
Adams, N. D.
Maynard, H. R.
Plummer, R. S.

St. Paul—
Anderson, D. B.
Ayers, E. H.
Backstrom, R. E.
Barnum, C. R.
Bauer, A. E.
Bean, G. S.
Cook, G. E.
Cuthbertson, M. W.
Diamond, D. D.
Estes, E. C.
Fitts, C. D.
Gausman, C. E.
Hickey, D. W.
Hyde, L. L.
Jones, E. F.
McNamara, W.
Mitchell, J. G.
Oberg, H. C.
Persson, N. B.
Ruff, D. C.
Sanford, A. L.
Swanstrom, A. E.
Winterer, F. C.
Wunderlich, M. S.

Wayzata—
Heberling, C. W.

MISSOURI

Clayton—
DuBois, L. J.

Ferguson—
Szombathy, L. R.

Independence—
Cook, B. F.

Jefferson City—
Woodman, L. E.

Kansas City—
Adams, C. W.
Allen, D. M.
Angus, F. M.
Arthur, J. M., Jr.
Ball, W.
Barnes, A. R.
Barnes, H. P.
Betz, H. D.
Bliss, G. L.
Caleb, D.
Cameron, W. R.
Campbell, E. K.
Campbell, E. K., Jr.
Carlson, C. V.
Case, D. V.
Cassell, W. L.
Chase, L. R.
Clegg, C.
Dawson, T. L.
Dean, F. J., Jr.
Dean, M. H.
DeVilbiss, P. T.
Disney, M. A.
Dodds, F. F.
Downes, N. W.
Farber, L. M.
Fehlig, J. B.
Flarsheim, C. A.
Forslund, O. A.
Garnett, R. E.
Gillham, W. E.
Gould, H. E.
Haas, E., Jr.
Hallar, E. V.
Harbordt, O. E.
Kitchen, J. H.
Lang, J. C.
Maillard, A. L.
Marston, A. D.
Matthews, J. E.
Middleton, H. A.
Millis, L. W.
Moore, B. J., Jr.
Natkin, B.
Nottberg, G.
Nottberg, H.
Nottberg, H., Jr.
Painter, D. H.
Pellmounter, T.
Pettit, E. N., Jr.
Pexton, F. S.
Pines, S.
Rivard, M. M.
Robb, J. E.
Sawyer, J. N.
Scarlett, W. J.
Schick, K. W.
Sheppard, F. A.
Stephenson, L. A.
Stevens, K. M.
Weiss, C. A.
White, H. S.
Wright, H. H.
Zink, D. D.

Kirkwood—
Hartwein, C. E.

Maplewood—
Droppers, C. J.
Siegel, W. A.

Mexico—
Badaracco, J. A.

Normandy—
Dulle, W. L.

Richmond Heights—
Nelson, C. L.

Springfield—
James, R. E.
Karchmer, J. H.

St. Joseph—
Flynn, F. J.
Harton, A. J.
Zurow, W.

St. Louis—
Allen, C. V.
Barry, J. G., Jr.
Bayse, H. V.
Boester, C. F., Jr.
Bradley, E. P.
Carlson, E. E.
Carter, J. H.
Cooper, J. W.
Corrigan, J. A.
Davis, C. R.
Driemeyer, R. C.
Edwards, D. F.
Evans, B. L.
Fagin, D. J.
Foster, J. M.
Gilmore, L. A.
Grossmann, H. A.
Haller, A. L.
Hamig, L. L.
Hamilton, J. E.
Hester, T. J.
Hugoniot, V. E.
Irwin, R. R.
Kent, J. K.
Kimmel, W. G.
Kuntz, E. C.
Langenberg, E. B.
Laufketter, F. C.
Lautz, F. A.
Malone, J. S.
Matousek, A. G.
McLarney, H. W.
McMahon, T. W.
Moon, L. W.
O'Brien, W. N.
Oonk, W. J.
Rickner, C. A.
Rodenheiser, G. B.
Rosebrough, J. S.
Rosebrough, R. M.
Scherrer, L. B.
Sodemann, P.
Sodemann, W. C. B.
Stammer, E. L.
Sydow, L. J.

Tenkonohy, R. J.
Warren, J. S., Jr.
Weber, E. F.
White, E. A.

University City—
Falvey, J. D.

Webster Groves—
Harbaugh, J. W.
Myers, G. W. F.
Ronsick, E. H.

MONTANA

Big Timber—
Strickland, A. W.

Billings—
Cohagen, C. C.
Young, F. H., Jr.

Great Falls—
Ginn, T. M.

NEBRASKA

Clarks—
Manning, W. M.

Columbus—
Ragatz, T. E.

Lincoln—
Darling, J. K.
Dexter, E. R.
Hellmers, C. C., Jr.
Hennessy, W. J.
King, L. D.
Leach, L. S.
Lehman, M. G.
Matthews, W. M.
Prawl, F. E.
Ress, O. J.
Rosenbach, R. F.
Shapiro, M. M.
Stanton, H. W.
Stevenson, M. J.
Tapley, M. S.
Williams, D. D.

Omaha—
Anderson, J. W.
Banner, F. L. D.
Goll, W. A.
Kleinkauf, H.
Larkin, P.
Lindberg, A. F.
Lycan, L. K.
Malcolm, B. L.
Moffitt, L. C.
Olson, M. J.
Peiser, M. B.
Rigby, R. A.
Rist, L. M.
Sallander, H. A.
Solzman, I. I.
Tracy, W. E.
White, W. R.

Scottsbluff—
Davis, O. E.

NEW HAMPSHIRE

Elkins—
Baker, R. H.

NEW JERSEY

Arlington—
Adler, A. A.
Bock, B. A.

Atlantic City—
Labov, M.
Strouse, S. B.

Asbury Park—
Strevell, R. P.

Bayonne—
Schwartz, J.

Belleville—
Thornton, T. L.

Bloomfield—
Clericuzio, G. P.
Faust, F. H.
Harrington, E.
McLenegan, D. W.
Tenney, D.

Bogota—
Griess, P. G.

Camden—
Brown, W. M.
Coward, C. W.
Kappel, G. W. A.
Lanning, E. K.
Plum, L. H.
Webster, E. K.
Webster, W.
Webster, W., Jr.

Clifton—
Hilder, F. L.

Cliffside Park—
Butler, P. D.

Collingswood—
Bolsinger, R. C.
Mohrfeld, H. H.

Cranford—
Roos, E. B. J.

Dover—
Hedden, W. M.

East Orange—
Atkins, T. J.
Ferguson, R. R.
Gombers, H. B.
Maddux, O. L.
Reilly, J. H.
Settelmeyer, J. T.
Tallmadge, W.
Turno, W. G. W.
Wadsworth, R. H.

Elizabeth—
Cornwall, G. I.
Faulkner, G.
Lyman, S. E.
Traynor, H. S.
Wheller, H. S.

Essex Fells—
Soule, L. C.
Stacey, A. E., Jr.

Freehold—
Buck, D. T.

Hasbrouck Heights—
Goodwin, S. L.

Haworth—
Sharp, J. R.

Hawthorne—
Spoelstra, W. J.

Hoboken—
Munson, J. L.

Irvington—
Reinke, A. G.
Stengel, F. J.

Jersey City—
Jones, H. L.
Ritchie, W.
Walterthum, J. J.

Lyndhurst—
Ehrlich, M. W.

Maplewood—
Kepler, D. A.
Kylberg, V. C.
Woolley, J. H.

Merchantville—
Binder, C. G.
Rohlin, K. W.

Montclair—
Bentz, H.

Newark—
Bryant, P. J.
Carey, P. C.
Holbrook, F. M.
Leinroth, J. P.
Morehouse, H. P.
Ray, L. B.
Raymer, W. F., Jr.
Steinmetz, C. W. A.

North Arlington—
Bermel, A. H.
Trambauer, C. W.

North Bergen—
Constance, J. D.

Orange—
Crawford, J. H., Jr.

Paterson—
Bannon, L. E.
Cox, H. F.
Pryor, F. L.

Perth Amboy—
Simkin, M.

Plainfield—
Tobin, G. J.

Ridgefield Park—
Davis, A. C.

Ridgewood—
Fitts, J. C.
Wallace, D. R.

Rochelle Park—
Emery, G. W.

Roselle—
Snavely, E. R.

Roselle Park—
Kampish, N. S.

Somerville—
Van Nuys, J. C.

South Orange—
Browne, A. L.
Feldermann, W.

Summit—
Oaks, O. O.

Teaneck—
Heebner, W. M.

Trenton—
Wagner, J. G.

Union City—
Taverna, F. F.

Upper Montclair—
Smith, F. J.

Verona—
Boxall, F.
Shotwell, R. W.

West Englewood—
Gumaer, P. W.
Steinke, B. J.

Westfield—
Scribner, E. D.

West Orange—
Adlam, T. N.

West New York—
Stinard, R. L.

NEW YORK

Albany—
Bond, H. A.
Dick, A. V.
Johnson, H. S.
Lewis, H. F.
Murray, T. F.
Nelson, A. W.
Ryan, H. J.
Taggart, R. C.
Teeling, G. A.
Westover, W.

Bedford Hills—
Mehne, C. A.
Waters, F. A.

Bronxville—
Bishop, C. R.
Dornheim, G. A.

Buffalo—
Beman, M. C.
Booth, C. A.
Bulkeley, C. A.
Cherry, L. A.
Cheyney, C. C.
Currier, C. H.
Curtis, W. A.
Davis, J.
Day, H. C.
Drake, G. M.
Eisele, W. S.
Farnham, R.
Farrar, C. W.
Gifford, C. A.
Grieves, T. R.
Harding, L. A.
Hart, F. D.
Hawk, J. K.
Heath, W. R.
Hedley, P. S.
Hexamer, H. D.
Hirschman, W. F.
Jackson, M. S.
Kaiser, F.
Kamman, A. R.
Landers, J. J.
Lenihan, W. O.
Lighthart, C. H.
Love, C. H.
Madison, R. D.
Mahoney, D. J.
Messenger, T. I.
Mosher, C. H.
Reif, A. F.
Reif, C. A.
Rente, H. W.
Roebuck, W., Jr.
Schafer, H. C.

Schmidt, H.
Seelbach, H.
Seelbach, H., Jr.
Sheiney, T.
Snyder, J. S.
Sweatt, C. H.
Vorsanet, W. E.
Walker, F. R.
Wendt, F. F.
Yager, J. J.

Corning —
Bates, H. C.

Croton —
Elliott, I.

Derby —
Ensign, W. A.

Elmira —
Davis, B. C.

Glens Falls —
Hollister, E. W.

Hamburg —
Graham, C. H.

Hartsdale —
Hedlund, R. A.

Hastings-on-Hudson
Reynolds, T. W.

Hyde Park —
Burr, G. C.

Irvington —
Bastedo, A. E.
Murphy, C. G.

Ithaca —
Barnes, A. A.
Elwood, W. H.
Fairbanks, F. L.
Sawdon, W. M.
Williams, J. W.

Kendall —
Stangland, B. F.

Kenmore —
Candee, B. C.
Crigui, A. A.
Mollenberg, H. J.
Summers, W. J.

Larchmont —
Downe, E. R.
Gaylord, W. S.

Lockport —
Prudden, O. D.
Saunders, L. P.

Montgomery —
Vogelbach, O.

Mt. Vernon —
Cribari, H. E.
Finnerty, J. A.
Freitag, F. G.
Northon, L.
Obert, C. W.

New Rochelle —
Abrams, A.
Farley, W. F.
Giannini, M. C.
Lambert, R. D.
Rose, H. J.
Terry, M. C.
Wallace, G. N.

New York City—
Addams, H.
Adler, J. C.
(Forest Hills, L. I.)
Alt, H. L.
(St. Albans)
Apt, S. R.
(Flushing)
Ashley, E. E., Jr.
Baker, H. L., Jr.
Ballman, W. H.
Balsam, C. P.
(Brooklyn)
Barbieri, P. J.
Barnum, M. C.
Bastedo, G. R.
(Richmond Hill)
Baum, A. L.
Bearman, A. A.
Beebe, F. E. W.
Belsky, G. A.
Bennett, E. A.
Berman, L. K.
Bernhard, G.
(Brooklyn)
Biancalli, V. A.
Blackburn, E. C., Jr.
(Garden City, L. I.)
Blackman, A. O.
Blackmore, J. J.
Blackshaw, J. L.
(Brooklyn)
Blakeslee, D.
(Patchogue)
Bloom, L.
(Brooklyn)
Bodinger, J. H.
Bolton, R. P.
Bonthron, R. C.
Borak, E.
Bowles, P.
Brabbee, C. W.
Broome, J. H.
(Brooklyn)
Brown, D.
Buenaod, A. C.
Burr, K.
Buttaravoli, F.
(Brooklyn)
Callahan, P. J.
(Great Kills, S. I.)
Campbell, F. B.
Campbell, R. E.
(Brooklyn)
Carbone, J. H.
(Baldwin, L. I.)
Carpenter, R. H.
Charles, T. J.
(Brooklyn)
Charlet, L. W.
Chase, C. L.
Connell, H.
(Staten Island)
Cross, F. G.
Cucci, V. J.
Dailey, J. A.
(Astoria, L. I.)
Daly, R. E.
Darts, J. A.
Davison, R. L.
Deely, J. J.
(Brooklyn)
Denny, H. R.
De Somma, A. E.
(Brooklyn)
Deterling, W. C.
Dodge, H. A.
Donnelly, R.
Downs, C. R.
Duff, K.
Duncan, J. R.
Dwyer, T. F.
(Brooklyn)
Eadie, J. G.
Elliott, L.
Engle, A.
Erickson, E. V.
Etlinger, M. J.
Everetts, J., Jr.
Faile, E. H.
Fay, D. P.
(Richmond Hill)
Fay, F. C.
Feldman, A. M.
Fenner, N. P.
Fidelius, W. R.
(Brooklyn)
Fiedler, H. W.
Fischer, L.
(Sayville, L. I.)
Fitz, J. C.
Fleisher, W. L.
Frank, O. E.
Friedman, M.
Frimet, M.
(Stapleton, S. I.)
Fritz, C. V.
(Freeport)
Galloway, J. F.
(Kew Gardens, L. I.)
Gates, R. A.
(Brooklyn)
Gitterman, H.
Glore, E. F.
Goldberg, M.
(Brooklyn)
Goldschmidt, O. E.
Gordon, P. B.
Gornston, M. H.
(Brooklyn)
Goulding, W.
(Brooklyn)
Graber, E.
(Astoria, L. I.)
Greben, D.
Green, A. W.
(Jackson Heights, L. I.)
Greenberg, I.
Greenburg, L.
Groves, S. A.
Hament, L.
Hamje, M. C.
(Brooklyn)
Harsch, R. J.
(Brooklyn)
Hateau, W. M.
Hechler, S.
Heibel, W.
Henry, A. S., Jr.
Hering, A.
Herkimer, H.
Herske, A. R.
Herty, F. B.
(Brooklyn)
Hiers, C. R.
(Great Neck)
Hildreth, L. W.
Hinkle, E. C.
(Hempstead)
Hinrichsen, A. F.
Hobbie, E. H.
Hoehl, E. R.
Hoffman, C. S.
Hollister, N. A.
(Brooklyn)
Honerkamp, F.
Hosking, H. L.
Hotchkiss, C. H. B.
Hyman, W. M.
d'Issertelle, H. G.
Jacobus, D. S.
Jalonack, I. G.
(Patchogue, L. I.)
James, J. W.
Janet, H. L.
(Brooklyn)
Jarcho, M. D.
Johnson, E. B.
W. New Brighton, S. I.)
Johnson, W. A.
Johnston, W. H.
Jordan, W. D.
Josephson, S.
(Brooklyn)
Kaczenski, C.
(Bridgehampton, L. I.)
Kastner, G. C.
Kelly, C. J.
Kenney, T. W.
Kern, J. F., Jr.
(Elmhurst)
Kessler, J.
Kimball, D. D.
Knopf, C.
(Brooklyn)
Koehler, C. S.
Kuhlmann, R.
Kurth, F. J.
Landewit, C. J.
(St. Albans, L. I.)
Lane, D. D.
(Elmhurst)
Leventhal, B.
(Brooklyn)
Lewis, C. A.
(Astoria, L. I.)
Lewis, T.
Lucke, C. E.
Lyons, M. A.
Maher, T. F., Jr.
(St. Albans, L. I.)
Markush, E. U.
Martens, E. D.
Martin, G. W.
Matzen, H. B.
(Rockville Centre, L. I.)
McClintock, W.
McCloughan, C.
(Brooklyn)
McEwan, E. E.
McGaughey, J. E. Jr.
McKiever, W. H.
McLeish, W. S.
Meinke, H. G.
Merle, A.
(Jackson Heights, L. I.)
Meyer, C. L.
(Hollis, L. I.)
Meyer, H. C., Jr.
Milener, E. D.
Miller, C. A.
Miller, J.
Montgomery, O. C.
Morse, F. W.
Moss, E.
(Brooklyn)
Munier, L. L.
Murphy, J. R.
Mytinger, K. L.
Neale, L. I.
Offner, A. J.
Oldes, W. E.
Olsen, G. E.
(Arverne, L. I.)
Olson, R. G.
Olvany, W. J.
Pabst, C. S.
(Woodhaven, L. I.)
Patorno, S. A. S.
Pfuhler, J. L.
(W. New Brighton, S. I.)
Phillips, F. W., Jr.
(Brooklyn)
Pietsch, J. A.
(New Brighton, S. I.)
Pihlman, A. A.
Pinto, C. B.
(Lawrence, L. I.)
Place, C. R.
Pohle, K.

Pollak, R.
Purdy, R. B.
Purinton, D. J.
Quirk, C. H.
Raisler, R. K.
Ramsay, J. W.
Rather, M. F.
Raynis, T.
(Richmond Hill)
Reynolds, W. V.
Richfield, N. H.
(Floral Park, L. I.)
Riley, R. C.
(Jamaica)
Ritchie, E. J.
(Brooklyn)
Ritter, A.
Rodman, R. W.
Rose, J. C.
(Brooklyn)
Rosenberg, P.
Rosenburg, W. E.
(Locust Valley, L. I.)
Rosenthal, E.
Ross, J. O.
Roth, C. F.
Roy, A. C.
Rudd, D. J.
(Babylon)
Ruggles, R. F.
(Randall Manor, S. I.)
Salter, E. H.
Samuels, S.
Sanbern, E. N.
(Rockville Centre, L. I.)
Sawhill, R. V.
Schermer, R.
(Elmhurst, L. I.)
Schoepflin, P. H.
Schulein, E. H.
(Forest Hills, L. I.)
Schulze, B. H.
Scott, A. F. H.
Scott, G. M.
Seelig, A. E.
Sellman, N. T.
Senior, R. L.
Shaffer, C. E.
Sherbrooke, W. A.
Siebs, C. T.
Simpson, A. M.
Sklenarik, L.
Smith, M. S.
Stalb, J. G.
(Brooklyn)
Steeneck, K. C.
(Bellaire, L. I.)
Stellwagen, F. G.
(Woodhaven)
Sternberg, E.
Sterne, C. M.
(Long Island City)
Still, F. R.
Stravitsch, J. J.
(Ozone Park, L. I.)
Strock, C.
Strunin, J.
Syska, A. G.
Taze, E. H.
Thomson, T. N.
(Huntington, L. I.)
Tiltz, B. E.
Timmis, W. W.
Torrance, H.
Tucker, F. N.
(Freeport, L. I.)
Tusch, W.
(Brooklyn)
Tyler, R. D.
Vervoort, E. L.
(Rockville Centre, L. I.)
Vetlesen, G. U.
Vivarttas, E. A.
(W. New Brighton, S. I.)
Vroome, A. E.
(Prince Bay, S. I.)
Wachs, L. J.
(Brooklyn)
Wade, R. H.
(Laurelton, L. I.)
Waechter, H. P.
(Staten Island)
Wagner, F. H., Jr.
Walker, W. K.
Wallace, G. J.
(East Elmhurst)
Walsh, M.
Walton, C. W., Jr.
Waring, J. M. S.
Wheeler, J., Jr.
Whitelaw, H. L.
Whittaker, W. K.
(St. Albans, L. I.)
Wilder, E. L.
Willner, I.
Wilson, A.
(Brooklyn)
Wilson, H. B.
(Brooklyn)
Wolf, P.
Wolin, M. W.
Wood, R. A.
(West New Brighton, S. I.)
Wrightson, W. T.

Niagara Falls—
Kessler, M. E.

North Pelham—
Dome, A. G.

North Tarrytown—
Weiss, A. P.

North Tonawanda—
Conaty, B. M.

Oswego—
Mohn, H. L.

Pelham—
Crone, T. E.

Pelham Manor—
Peacock, J. K.

Rochester—
Andresen, G. C.
Betlem, H. T.
Cook, R. P.
Eschenbach, S. P.
Hakes, L. M.
Hewett, J. B.
Hutchins, W. H.
Lee, R. T.
Leonhard, L. W.
Stacy, S. C.
Treadway, J. Q.
Vidale, R.

Rome—
Lynch, W. L.

Scarsdale—
Cumming, R. W.
Ullman, H. G.

Schenectady—
Danielson, L. C.
Hunziker, C. E.
Seely, I. R.
Vogel, A.
Welch, L. A., Jr.

Schodack Landing—
Freas, R. B.

Snyder—
John, V. P.

Syracuse—
Acheson, A. R.
Ashley, C. M.
Cady, E. F.
Carrier, W. H.
Cherne, R. E.
Day, V. S.
Dee, L. H.
Des Reis, J. F.
Driscoll, W. H.
Evans, E. C.
French, D.
Graham, W. D.
Grant, W. A.
Hockensmith, F. E.
Ingels, M. M.
Jackes, H. D.
Lewis, L. L.
Lyle, J. I.
O'Rourke, H. D., Jr.
Perina, A. E.
Schulz, E. L.
Sheldon, N. E.
Taliaferro, R. R.
Woese, C. F.

Tonawanda—
Karlsteen, G. H.

Utica—
Knapp, J. H.
Price, E. H.
Skidmore, J. G.
Steinhorst, T. F.

White Plains—
Bennitt, G. E.
Durkee, M. E.
Rose, A. A.
Ruple, P. E.
Zibold, C. E.

Yonkers—
Dean, D.
Goerg, B.
Harmonay, W. L.
Hayter, B.
Hopp, H. K.
Kelly, J. G.
Rainger, W. F.
Zuhlke, W. R.

NORTH CAROLINA

Charlotte—
Arden, I. L.
Brandt, E. H., Jr.
Campbell, A. Q., Jr.
Hill, H. H.
Hodge, W. B.
Muirheid, J. G.
Shirley, W. B.

Durham—
Cooke, T. C.

Greensboro—
Harding, E. R.
Small, B. R.

High Point—
Gray, W. E.

Raleigh—
Rice, R. B.

Winston-Salem—
Bahnson, F. F.
Brown, M. D.
Cornwall, C. C.
Page, A.

OHIO

Akron—
McElhaney, G. W.
Shields, C. D.

Cincinnati—
Bird, C.
Boyd, T. D.
Coombe, J.
Donelson, W. N.
Doyle, W. J.
Edwards, A. W.
Fenker, C. M.
Green, W. C.
Helburn, I. B.
Houlis, L. D.
Houliston, G. B.
Hudepohl, L. F.
Hust, C. E.
Junker, W. H.
Kiefer, C. J.
Kinney, A. M.
Kramig, R. E., Jr.
Little, K. B.
Mason, G. C.
Mathewson, M. E.
Mayer, R. W.
Mayne, W. L.
Mills, C. A.
Pillen, H. A.
Pistler, W. C.
Powers, L. G.
Richard, E. J.
Royer, E. B.
Ruff, A. G.
Sigmund, R. W.
Silberstein, B. G.
Smith, J. A.
Smith, S.
Sproull, H. E.
Sutfin, G. V.
Washburn, M. J.
Winther, A.
Wright, K. A.

Cleveland—
Avery, L. T.
Beach, W. R.
Borkat, P.
Brezina, E. A.
Cary, E. B.
Cheeseman, E. W.
Cohen, H.
Cohen, P.
Conner, R. M.
Curtis, H. F.
Eveleth, C. F.
Friedman, A.
Geissbuhler, J. O.
Gottwald, C.
Gray, E. W.
Greenleaf, R. P.
Harris, J. G.
Harvey, L. C.
Heisterkamp, H. W.
Jones, J. P.
Kaercher, C. M. H.
Kain, E. M.
Kalinsky, A. G.
Kitchen, F. A.
Klie, W.
Kofoed, V. B.
Levy, M. I.
Marotta, J. A.
Martinka, P. D.
Maurer, F. J.
McKeeman, C. A.
Moeller, R.
Moore, W. R.
Pogalies, L. H.
Repko, J. J.
Rhoton, W. R.
Rowe, W. M.
Schurman, J. A.

Simon, A.
Smith, W. D.
Southmayd, R. T.
Taze, D. L.
Tuve, G. L.
Vanderhoof, A. L.
Walker, J. E.
Weatherby, E. P., Jr.
Webb, E. C.
Weldy, L. O.
Wetzell, H. E.
Wilhelm, J. E.
Wootan, C. C.
Young, E. O.

Cleveland Heights—
Davis, R. G.
Richmond, J.
Sugg, A.
Sterling, J. G., Jr.

Columbus—
Allonier, H. R.
Breneman, R. B.
Brown, A. I.
Cooper, W. B.
Cross, R. C.
Denise, J. R.
Kimmell, P. M.
Myler, W. M., Jr.
Poling, D. B.
Sherman, R. A.
Sidwell, E. W.
White, E. D.
Williams, A. W.
Wyatt, D. H.

Cuyahoga Falls—
Humphrey, D. E.
Jennings, H. K.

Dayton—
Baker, I. C.
Brown, J. S., Jr.
Buenger, A.
Chapman, W. A., Jr.
Clarke, J. G.
Gibbons, M. J., Jr.
Gonzalez, R. A.
Hander, J. E.
Hull, H. B.
Hutchings, R. L.
Kuempel, L. L.
La Salvia, J. J.
Lewis, C. E.
Livar, A. P.
MacMillan, A. R.
Newell, C. W.
Noble, J. P.
Sidell, P. A.
Snyder, A. K.
Swift, P. F.
Williams, F. H.
Wood, C. E.
Wordham, H.
Wyld, R. G.

East Cleveland—
Geltz, R. W.
Morris, F. H.
Nobis, H. M.
Sennet, L. E.
Stark, W. E.
Steffner, E. F.

Elyria—
Maynard, J. E.

Hamilton—
Thomas, L. G. L.

Hudson—
Follett, T. L.

Lakewood—
Kubasta, R. W.
Longcoy, G. B.
Tanker, G. E.
Teckmyer, F. C., Jr.
Wilson, R. A.

Middletown—
Byrd, T.
Maier, A. H.
Stitt, A. B.

Newark—
Simison, A. L.
Slayter, G.

Norwood—
Braun, J. J.
Motz, O. W.

Oberlin—
Ries, L. S.

Painesville—
Hobbs, J. C.

Parma—
Kajuk, A. E.

Piqua—
Lange, R. T.

Sandusky—
Rathke, A. C.

Shaker Heights—
Harvey, R. A.

South Park—
Barney, W. E.

Springfield—
Hauck, E. L.

Toledo—
Baker, H. C.
Bergan, J. R.
Damm, D. A.
Jones, S.
Mayes, C.
McKitrick, W. D.
Myers, F. L.
Watkins, G. B.

Waverly—
Armbruster, F. T. W.

Wilmington—
Marconett, V. G.
Sapp, C. L.

Worthington—
Slemmons, J. D.

Youngstown—
Boucherle, H. N.
Montgomery, J. R.

OKLAHOMA

Alva—
Husky, S. T.

Bartlesville—
Yates, G. L.

Norman—
Adams, B. C., Jr.
Dawson, E. F.
Giles, J. C.
Sonney, K. J.

Oklahoma City—
Carnahan, J. H.
Dolan, R. G.
Dugger, E. R.
Earl, W.
Gray, E. W.
Holyfield, E. F.
Hoppe, A. A.
Howlett, I. G.
Loeffler, F. X.
Loeffler, L., Jr.
Meinholtz, H. W.
Middleton, D. K.
Mideke, J. M.
Miller, B. R.
Patton, R. L.
Rathbun, P. W.
Rolland, S. L.
Spencer, D.
Tiller, L.

Tulsa—
Dean, C. H.
Holmes, A. D.
Jones, E.
Jordan, R. C.
Pauling, R. E.
Scherrer, K. C.
Shoemaker, F. F.

OREGON

Corvallis—
Willey, E. C.

Portland—
Kroecker, J. D.
Taylor, T. E.

PENNSYLVANIA

Abington—
Park, N. W.

Allentown—
Hersh, F. C.
Korn, C. B.

Ambler—
McElgin, J. W.

Ardmore—
Haynes, C. V.

Ardsley—
Tucker, L. A.

Bala-Cynwyd—
Patrick, H. M.
Whitney, C. W.

Beaver Falls—
Van Alen, W. T.

Bethlehem—
Bornstein, W.
Murnin, E. A., Jr.
Stuart, M. C.

Bradford—
Cleveland, C. C.
Paterson, F. C., Jr.
Presdee, C. W.
Turner, P. K.

Brookline—
Donnelly, M. A.

Clearfield—
Gault, G. W.

Dunbar—
Sherwood, L. T.

Elizabethtown—
Dibble, S. E.

Erie—
Joyce, H. B.
McDowell, B. W.
Preece, L. W.
Sahlmann, F. L.

Etna—
Park, H. E.

Glenside—
Werner, J. G.

Greensburg—
Burkhart, E. M.

Harrisburg—
Eicher, H. C.
Geiger, I. H.

Haverford—
Black, E. N., III

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Snavely, A. B.

Johnstown—
Huettner, H. F.
Hunter, L. N.
Knowles, F. R.
Novotney, T. A.

Kingston—
Macdonald, D. B.

Lancaster—
Chenoweth, D. M.
Jones, A.
Lloyd, E. C.

Lansdowne—
Hicks, H. K.
James, H. R.
Lauer, R. F.
Mawby, P.

Lemont—
Henszey, W. P.

Manheim—
Weitzel, C. B.
Weitzel, P. H.

McKeesport—
Dugan, T. M.

McKees Rocks—
Rodgers, F. E.

Middletown—
Locke, R. A.

Midland—
Crichton, H. C.

Narberth—
Dever, H. F.
Searle, W. J., Jr.

New Castle—
Andrews, G. H.
Sonneborn, C.

New Kensington—
Edwards, J. D.

Norristown—
Hucker, J. H.

Oxford—
Ware, J. H., III

Philadelphia—
Adams, B.
Ahlff, A. A.
Arnold, R. S.
Bachman, F.
Barnard, M. E.
Bartlett, C. E.
Black, H. G.
Blankin, M. F.
Bogaty, H. S.
Bornemann, W. A.
Burke, J.
Caldwell, A. C.
Call, J.
Cassell, J. D.
Clodfelter, J. L.
Cody, H. C.
Culbert, W. P.
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Davidson, L. C.
Dietz, C. F.
Donovan, W. J.
Elliot, E.
Erickson, H. H.
Evleth, E. B.
Faltenbacher, H. J.
Familetti, A. R.
Galligan, A. B.
Gant, H. P.
Gillett, M. C.
Gilman, F. W.
Guler, G. D.
Hackett, H. B.
Hance, W. W.
Hedges, H. B.
Hibbs, F. C.
Hunger, R. F.
Hynes, L. P.
Ickeringill, J.
Jopson, J. M.
Kelble, F. R.
Kriebel, A. E.
Landau, M.
Leopold, C. S.
Lyon, P. S.
MacDade, A. H.
Mack, L.
Martocello, J. A.
Mather, H. H.
McClintock, A., Jr.
McCullough, H. G.
McIlvaine, J. H.
Mellon, J. T. J.
Meloney, E. J.
Mensing, F. D.
Millham, F. B.
Moody, L. E.
Morgan, R. C.
Morris, A. M.
Munro, G. A.
Murdoch, J. P., Jr.
Nesbitt, A. J.
Nesbitt, J. J.
Newcomb, L. B.
Nusbaum, L.
Plewes, S. E.
Prylbil, P. L.
Rank, A. I.
Redstone, A. L.
Reilly, C. E.
Rettew, H. F.
Rhea, C. A.
Roberts, H. L.
Rugart, K.
Sabin, E. R.
Schneider, C. H.
Shanklin, A. P.
Sheffler, M.
Skagerberg, R.
Smith, D. K.
Smith, W. F.
Speckman, C. H.
Stewart, J. P.
Stokes, A. D.
Timmis, P.
Touton, R. D.
Traugott, M.
Trump, C. C.
Tuckerman, G. E.
Wegmann, A.
Wiley, D. C.
Wilmot, C. S.
Wiltberger, C. F.
Woolston, A. H.
Yerkes, W. L.

Pittsburgh—
Beighel, H. A.
Biber, H. A.
Blackmore, G. C.
Brauer, R.
Bushnell, C. D.
Carr, M. L.
Collins, J. F. S., Jr.
Comstock, G. M.
Dorfan, M. I.
Edwards, P. A.
Eils, L. C.
Ellis, G. P.
Frisse, J. L.
Giles, A. F.
Goodman, D. J.
Greiner, G. E., Jr.
Griest, K.
Hecht, F. H.
Heilman, R. H.
Houghten, F. C.
Humphreys, C. M.
Hyde, E. H.
Kennedy, O. A.
Kennedy, P. V.
Landes, B. D.
Loucks, D. W.
Maehling, L. S.
Maginn, P. F.
Mahon, F. B.
Marshall, A. W.
McGonagle, A.
McGuigan, L. A.
McIntosh, F. C.
McLean, J. E.
Miller, R. A.
Moore, H. L.
Mueller, J. E.
Nass, A. F.
Neis, W. A.
Nicholls, P.
Parks, C. E.
Proie, J.
Reed, I. G.
Reed, V. A., Jr.
Riesmeyer, E. H., Jr.
Rockwell, T. F.
Rodgers, W. C.
Rose, H. J.
Scanlon, E. S.
Selig, E. T., Jr.
Smyers, E. C.
Speller, F. N.
Stanger, R. B.
Steggall, H. B.
Stevenson, W. W.
Strauch, P. C.
Tennant, R. J. J.
Tower, E. S.
Waters, G. G.
Weddell, G. O.
Zangrilli, A. J.

Pottsville—
Marty, E. O.
Smith, J. D.

Primos—
Hall, M. S.
Johnson, A. J.

Radnor—
Davidson, P. L.

Reading—
Luck, A. W.
Nicely, J. E.

Rutledge—
Jones, N. R.

Scranton—
Mahon, B. B.
Shaver, H. H.

Springfield—
Grossman, H. E.
Payne, R. E.

State College—
Queer, E. R.

Stroudsburg—
Kiefer, E. J., Jr.

Swarthmore—
Hobbs, W. S.
Robinson, A. S.
Thom, G. B.

Tarentum—
Orr, L.

Uniontown—
Marks, A. A.

Upper Darby—
Aughenbaugh, H. E.
Beitzell, A. E.
Blackmore, J. S.
Currie, F. J.
Eastman, C. B.
Kipe, J. M.
McClain, C. H.

Villanova—
Barr, G. W.
Carey, J. A.

Washington—
Frazier, J. E.

Wilkinsburg—
Campbell, T. F.

Williamsport—
Axeman, J. E.

Wormleysburg—
Miller, T. G.

Wyncote—
Buck, L.

York—
Barnum, W. E., Jr.
Clippinger, J. V.
Hertzler, J. R.
Kartorie, V. T.
Walsh, E. R., Jr.
Zieber, W. E.

PHILLIPINE ISLANDS

Manila—
Hausman, L. M.
Macrae, R. B.

RHODE ISLAND

Providence—
Coleman, J. B.
Gibbs, E. W.
Hartwell, J. C.
McCarthy, J. J.
McLaughlin, J. D.
Moulder, A. W.

SOUTH CAROLINA

Clemson College—
Shenk, D. H.

Columbia—
Hartin, W. R., Jr.
Kerr, W. E.
Mercer, C. F.
Reamer, W. S., Jr.

SOUTH DAKOTA

Lead—
Pullen, R. R.

Sioux Falls—
Monick, F. R.

TENNESSEE

Chattanooga—
Campbell, G. S.

Elizabethton—
Torok, E.

Knoxville—
Oakley, L. W.

Memphis—
Flinn, G. S.
Hoshall, R. H.

Murfreesboro—
Armistead, W. C.
Moore, H. W.

Nashville—
Brown, F.

TEXAS

Amarillo—
Burnett, E. S.
Houska, A. D.

Bryan—

Griesser, C. E.
Rogers, R. C.

College Station—

Badgett, W. H.
Giesecke, F. E.
Hines, G. M.
Long, W. E.
Smith, F. G.
Teal, E. T.

Dallas—

Anspacher, T. H.
Bock, I. I.
Boruch, E. R.
Constant, E. S.
Durning, E. H.
Gardner, C. R.
Gilbert, L. S.
Jelinek, F. R.
Kribs, C. L., Jr.
Landauer, L. L.
Martyn, H. J.
Mehl, O. H.
Moler, W. H.
Renouf, E. P.
Schmidt, H. I.
Schucany, O. W.
Stringfellow, J. C.
Turner, H. S., Jr.
Ullrich, A. B.
Witmer, C. N.

El Paso—

Rodgers, F. A.

Fort Worth—

Skinner, H. W.
Werner, R. K.

Galveston—

Huff, J. M.
Warren, C. W.

Hereford—

Jones, A. P.

Houston—

Cheatwood, W. H.
Cochran, W. B.
Cooper, D. S.
Kiesling, J. A.
Kurtz, R. W.
McKinney, C. A.
Rowe, I. E.
Salinger, R. J.
Taylor, R. F.
Walsh, J. A.

Irving—

Wilson, J. W.

Kingsville—

Richtmann, W. M.

Port Arthur—

Shaw, C. G.

San Antonio—

Billingsley, O. F.
Cotter, R. P.
Diver, M. L.
Ebert, W. A.
Kotzebue, R. W.
Monier, K. A. J.
Rummel, A. J.

Somerville—

Barton, D. H.

UTAH

Salt Lake City—

Richardson, H. G.
Young, J. T., Jr.

VERMONT

Burlington—

Lanou, J. E.
Raine, J. J.

North Ferrisburg—

Breckenridge, L. P.

VIRGINIA

Arlington—

Ferrarini, J.
Marshall, W. D.
Rudio, H. M.
Welsh, H. A.

Blacksburg—

Johnston, R. M.

Charlottesville—

Peebles, J. K., Jr.

Lynchburg—

Doering, F. L.

Norfolk—

Capps, E. L.
Nowitzky, H. S.
Webster, W. H., Jr.

Portsmouth—

Stubbs, W. C.

Richmond—

Belding, H. H.
Carle, W. E.
Johnston, J. A.
Pelouze, H. L., II
Schulz, H. I.

WASHINGTON

Kent—

Boyker, R. O.

Port Orchard—

Pratt, F. J.

Seattle—

Beggs, W. E.
Bouillon, L.
Case, R. H.
Cox, W. W.
Daly, C. P.
Eastwood, E. O.
Granston, R. O.
Hauan, M. J.
MacLeod, K. F.
Mallis, W.
May, C. W.
Morse, R. D.
Musgrave, M. N.
O'Connell, P. M.
Peterson, S. D.
Pollard, A. L.
Sparks, J. D.
Twist, C. F.
Ward, J. J.
Watt, R. D.
Weber, E. L.
Wesley, R. O.
Zokelt, C. G.

Spokane—

Russell, W. B.

Tacoma—

Foote, E. E.
Spofforth, W.

Yakima—

Leichnitz, R. W.
McCune, B. V.

WEST VIRGINIA

Charleston—

Rothmann, S. C.
Shanklin, J. A.
Wright, M. B.

Fairmont—

Tonry, R. C.
Wright, C. E.

Huntington—

Johnson, L. O.

Largent—

Donnelly, J. A.

Wheeling—

Hitt, J. C.

WISCONSIN

Beloit—

McKinley, C. B.

Clintonville—

Quall, C. O.

Cuba City—

Kellner, D. C.

Green Bay—

Haus, I. J.

Kohler—

Hvoslef, F. W.
Kohler, W. J., Jr.

La Crosse—

Anderegg, R. H.
Bledsoe, R. P.
Miller, M. W.
Rowe, W. A.
Thomas, N. A.
Trane, R. N.

Madison—

Dean, C. L.
Feirn, W. H.
Hall, G.
Larson, G. L.
Nelson, D. W.
Seymour, J. E.
White, J. C.

Milwaukee—

Allan, W.
Banks, J. B.
Berghoefer, V. A.
Bernert, L. A.
Boden, W. F.
Bowers, A. F.
Brown, W. H.
Cook, H. R.
Davis, K. T.
Elliott, N. B.
Ellis, H. W.
Errath, E. O.
Frentzel, H. C.
Goldsmith, F. W.
Gregg, S. H.
Hanley, E. V.
Hays, C. A.
Hessler, L. W.
Hughey, T. M.
Jackson, C. H.
Jepertinger, R. C.
Jones, E. A.
Jung, J. S.
Ketter, J. W.
Knab, E. A.
Koch, R. G.
Krenz, A. S.
Lofte, J. A.
Miller, C. W.
Mueller, H. P.
Noll, W. F.
Page, H. W.
Randolph, C. H.
Reinke, L. F.
Rice, C. J.
Schreiber, H. W.
Shawlin, W. C.
Shodron, J. G.
Sickert, G. D.
Spence, M. R.
Steinkellner, E. J.
Swisher, S. G., Jr.
Szekely, E.
Trostel, O. A.
Volk, J. H.
Wagner, A. M.
Weimer, F. G.
Winkler, R. A.

Neenah—

Angermeyer, A. H.
Eiss, R. M.
Stiegler, A. J.

Racine—

Dixon, A. G.
Menden, P. J.

South Milwaukee—

Ouweneel, W. A.

Superior—

Waite, H.

West Allis—

Erickson, M. E.
Spence, R. T.
Wieriman, W. J.

CANADA

Brandon, Man.—
Yates, J. E.
Yates, J. E., Jr.

Calgary, Alberta—
Vissac, G. A.

Edmonton, Alberta—
Mould, D. E.

Flin Flon, Man.—
Foster, P. H.

Fort Garry, Man.—
Davis, G. C.

Freeman, Ont.—
Goodram, W. E.

Galt, Ont.—
Sheldon, W. D., Jr.

Hamilton, Ont.—
Barnes, H.
Best, M. W.
Charters, W. A.
Dickenson, M. E.
Moffat, O. G.

Islington, Ont.—
Wilson, G. T.

Kirkland Lake, Ont.—
Calver, R. W.

Kitchener, Ont.—
Beavers, G. R.
Pollock, C. A.

Lindsay, Ont.—
McCrae, G. W.

London, Ont.—
Dobie, T. K.
O'Flaherty, J. G.

Montreal, P. Q.—
Allaire, L.
Ballantyne, G. L.
Barnsley, F. R.
Darling, A. B.
Dewar, W. G.
Dixon, M. F.
Dufault, F. H.
Dupuis, J. R.
Dykes, J. B.
Forrester, N. J.
Freeman, E. M.
Friedman, F. J.
Garneau, L.
Gendron, H.
Gittleson, H.
Hamlet, A.
Hughes, W. U.
Johnson, C. W.
Laffoley, L. H.
Lamontagne, A. F.
Linton, J. P.
Madely, F. J.
Marshall, A. G.
Martin, L.
Martin, R.
Morris, J. A.
Murray, H. G. S.
Nickle, A. J.
Osborne, G. H.
Peart, A. M.
Perras, G. E.
Phipps, F. G.
Robertson, J. A. M.
Roche, I. F.
Ross, J. D.
Timmins, W. W.
Tolhurst, G. C.
Twizell, E. W.
Watts, A. E.
Wiggs, G. L.
Wilkinson, A.
Wilson, A.
Worthington, T. H.

Ottawa, Ont.—
Allen, A. W.
Colclough, O. T.
Gray, G. A.
McGrail, T. E.
Pennock, W. B.

Preston, Ont.—
Everest, R. H.

Quebec, Que.—
La Rocque, P. E.
Paquet, J. M.
Roy, L.

Rigaud, Que.—
Fogarty, O. A.

St. Catherines, Ont.—
Palmason, J. H.

St. Lambert, Que.—
Lefebvre, E. J.

Three Rivers, Que.—
Germain, O.

Timmins, Ont.—
Smith, R. J.

Toronto, Ont.—
Alexander, S. W.
Allcut, E. A.
Allsop, R. P.
Angus, H. H.
Anthes, L.
Arrowsmith, J. O.
Baker, G. R.
Baker, L. P.
Blackhall, L. C.
Blackhall, W. R.
Boddington, W. P.
Bowerman, E. L.
Cairns, J. H.
Carter, A. W.
Chambers, F. W.
Church, H. J.
Cole, G. E.
Cornish, D. F.
Davenport, R. F.
Dickey, A. J.
Dion, A. M.
Dowler, E. A.
Duncan, W. A.
Eaton, W. G. M.
Ellis, F. E.
Ewens, F. G.
Fitzsimons, J. P.
Forrester, C. M.
Fox, E.
Fox, J. H.
Gauley, E. R.
Givin, A. W.
Gordon, W. D.
Gurney, E. H.
Gurney, E. R.
Harrington, C.
Heard, R. G.
Henion, H. D.
Hills, A. H.
Hopper, G. H.
Hughes, L. K.
Jeffrey, T. G.
Jenney, H. B.
Jennings, S. A.
Jones, A. T.
Kelly, W. C.
Lawlor, J. J.
Ledgett, F. D.
Leitch, A. S.
Libby, R. S.
Lower, H. C.
MacDonald, D. J.
Marriner, J. M. S.
Maxwell, R. S.
McCrimmon, A. M.
McDonald, T.
Moore, H. S.
Oke, W. C.
O'Neill, J. W.
Paul, D. I.
Philip, W.
Playfair, G. A.
Price, D. O.
Ritchie, A. G.
Roth, H. R.
Shears, M. W.
Sheppard, W. G. F.
Stott, F. W.
Tasker, C.
Thomas, M. F.
Waldon, C. D.
Wardell, A.
Watson, M. B.
Whittall, E. T.
Willer, M. D.
Woollard, M. S.
World, H. P.

Vancouver, B. C.—
Dawson, G. S.
Hale, F. J.
Johnston, R. E.
Leek, W.
McCreery, H. J.
Turland, C. H.

Victoria, B. C.—
Sheret, A.

Wellington, Ont.—
Johnston, H. D.

Westmount, Que.—
Colford, J.
Pratt, J. C.

Windsor, Ont.—
Aitken, J.
Hare, W. A.

Winnipeg, Man.—
Argue, E. J.
Cunliffe, J. A.
Eade, H. R.
Glass, W.
Jones, B. G.
Kent, R. L.
Kipp, T.
Michie, D. F.
Miller, E. R.
Munn, E. F.
Steele, J. B.
Summers, E. T.
Thompson, F.
Watson, H. D.
Worton, W.

Woodstock, Ont.—
Karges, A.

AUSTRALIA

Melbourne—
Atherton, A. E.
Ross, R.

Sydney—
Davey, G. I.
Hunt, N. P.
Moloney, R. R.
Picot, J. W.
Robinson, J. A.
Sands, C. C.

BELGIUM

Brussels—
Mautsch, R.

BRAZIL

Rio de Janeiro—
Botelho, N. J.
Darby, M. H.

CHINA

Nanking—
Loo, P. Y.

Shanghai—
Bradford, G. G.
Chen, S. T.
Doughty, C. J.
Gange, F. B.
Hart-Baker, H. W.
Kwan, I. K.
Loh, N. S.
Morrison, C. B.
Rachal, J. M.
Waung, T. F.

CUBA

Havana—
Edwards, H. B.

CZECHOSLOVAKIA

Praha—
Brust, O.

DENMARK

Copenhagen—
Reck, W. E.

DUTCH EAST INDIES JAVA

Soerabaia—
Thornburg, H. A.

ENGLAND

Birmingham—
Bird, G. L. H.

Caterham—
Carter, D.

Cheshire—
Adshead, B.

Dartford—
Figgis, T. G.

Denton—
Webb, J. W.

Leeds—
Jennins, H. H.

London—
Bailey, W. M.
Benham, C. S. K.
Butt, R. E. W.
Chester, T.
Faber, O.
Fraser, J. J.
Greenland, S. F.
Haden, G. N.
Herring, E.
Kraminsky, V.
Linebaugh, J. E.
Nobbs, W. W.
Pryke, J. K. M.
Russell, J. N.

Middlesex—
Case, W. G.
Gill, E. F.

Sutton—
Casperd, H. W. H.

Swinton—
Yates, W.

Trowbridge—
Haden, W. N.

Wolverhampton—
Tyson, W. H.

FRANCE

Dijon—
Bur, J. R. C.

Lille—
Neu, H. J. E.

Lyon—
Goenaga, R.

Paris—
Beaurrienne, A.
Bodmer, E.
Downe, H. S.
Ghilardi, F.
Modiano, R.
Nessi, A.
Schmutz, J.

GERMANY

Hamburg—
Brandi, O. H.

Stuttgart—
Klein, A. R.

INDIA

Bombay—
Wilson, E. D.

New Delhi—
Heard, J. A. E.

IRELAND

Cork—
Barry, P. I.

Dublin—
Leonard, L. C. G.

ITALY

Milan—
Gini, A.
Hauss, C. F.

Torino—
Baldi, G.

JAPAN

Osaka—
Fukui, K.

Tokyo—
Kitaura, S.
Kozu, T.
Saito, S.
Sekido, K.

MANCHOUKUO

Hsiking—
Kawase, S.

MEXICO

Mexico, D. F.
Gilfrin, G. F.
Martinez, J. J.

NETHERLANDS

Arnhem—
Tanger, O. C. F.

NEW ZEALAND

Christchurch—
Taylor, E. M.
Vale, H. A. L.

Dunedin—
Davies, G. W.

NORWAY

Oslo—
Alfsen, N.
Tjersland, A.

SCOTLAND

Angus—
Knox, J. R.

SOUTH AFRICA

Durban—
Kothe, F. H.

Johannesburg—
Carrier, E. G.
Ehlers, J.
Overton, S. H.
vonChristierson, C. A.

SPAIN

Madrid—
Alfageme, B.
Jimenez, J. G.

STRAITS SETTLEMENTS

Singapore—
Faxon, H. C.
Hill, C. F.

SWEDEN

Lidingo—
Rosell, A. F.

Stockholm—
Gille, H.
Ostrom, E. W.
Theorell, H. G. T.

TRINIDAD

Port of Spain—
Cox, T. M., Jr.

TURKEY

Istanbul—
Karakash, T. J.

U.S.S.R.

Moscow—
Dunne, R. V.

VENEZUELA

Caracas—
Blas, R. J.

UNITED STATES AND ISLAND TERRITORIES

Alabama	8	Indiana	31	Nebraska	36	South Dakota	2
Arizona	2	Iowa	10	New Hampshire	1	Tennessee	8
Arkansas	3	Kansas	11	New Jersey	98	Texas	57
California	91	Kentucky	19	New York	442	Utah	2
Colorado	8	Louisiana	14	North Carolina	16	Vermont	3
Connecticut	32	Maine	5	Ohio	179	Virginia	16
Delaware	9	Maryland	32	Oklahoma	32	Washington	30
Dis. of Columbia	56	Massachusetts	115	Oregon	3	West Virginia	8
Florida	12	Michigan	154	Pennsylvania	251	Wisconsin	77
Georgia	27	Minnesota	105	Phillipine Is.	2		
Hawaii	2	Missouri	132	Rhode Island	6		
Illinois	271	Montana	4	South Carolina	5		2427

DOMINION OF CANADA 182

FOREIGN COUNTRIES

Australia	8	France	10	Mexico	2	Sweden	4
Belgium	1	Germany	2	Netherlands	1	Trinidad	1
Brazil	2	India	2	New Zealand	3	Turkey	1
China	11	Ireland	2	Norway	2	U. S. S. R.	1
Cuba	1	Italy	3	Scotland	1	Venezuela	1
Czechoslovakia	1	Japan	5	South Africa	5		
Denmark	1	Java	1	Spain	2		103
England	26	Manchouko	1	Str't. Set'ments.	2		

Total Membership .. 2712

PAST OFFICERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1894

President	Edward P. Bates
1st Vice-President	Wm. M. Mackay
2nd Vice-President	Wiltsie F. Wolfe
3rd Vice-President	Chas. S. Onderdonk
Treasurer	Judson A. Goodrich
Secretary	L. H. Hart

Board of Managers

Chairman, Fred P. Smith

Henry Adams	A. A. Cary
Hugh J. Barron	James A. Harding
Edward P. Bates, *Pres.*	L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter

Albert A. Cryer	Chas. W. Newton
F. W. Foster	Ulysses G. Scollay, *Secy.*

1895

President	Stewart A. Jellett
1st Vice-President	Wm. M. Mackay
2nd Vice-President	Chas. S. Onderdonk
3rd Vice-President	D. M. Quay
Treasurer	Judson A. Goodrich
Secretary	L. H. Hart

Board of Managers

Chairman, James A. Harding

Geo. B. Cobb	Ulysses G. Scollay
Wm. McMannis	B. F. Stangland
Stewart A. Jellett, *Pres.*	L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams	T. J. Waters
Edward P. Bates	Albert A. Cryer, *Secy.*

1896

President	R. C. Carpenter
1st Vice-President	D. M. Quay
2nd Vice-President	Edward P. Bates
3rd Vice-President	F. W. Foster
Treasurer	Judson A. Goodrich
Secretary	L. H. Hart

Board of Managers

Chairman, Wm. M. Mackay

Hugh J. Barron	Stewart A. Jellett
W. S. Hadaway, Jr.	Wiltsie F. Wolfe
R. C. Carpenter, *Pres.*	L. H. Hart, *Secy.*

Council

Chairman, A. A. Cary

Albert A. Cryer	B. F. Stangland
Wm. McMannis	J. J. Blackmore, *Secy.*

1897

President	Wm. M. Mackay
1st Vice-President	H. D. Crane
2nd Vice-President	Henry Adams
3rd Vice-President	A. E. Kenrick
Treasurer	Judson A. Goodrich
Secretary	H. M. Swetland

Board of Managers

Chairman, R. C. Carpenter

Edward P. Bates	Stewart A. Jellett
W. S. Hadaway, Jr.	Wiltsie F. Wolfe
Wm. M. Mackay, *Pres.*	H. M. Swetland, *Secy.*

Council

Chairman, Albert A. Cryer

John A. Fish	James Mackay
Wm. McMannis	B. F. Stangland

1898

President	Wiltsie F. Wolfe
1st Vice-President	J. H. Kinealy
2nd Vice-President	A. E. Kenrick
3rd Vice-President	John A. Fish
Treasurer	Judson A. Goodrich
Secretary	Stewart A. Jellett

Board of Managers

Chairman, Wm. M. Mackay

Thomas Barwick	A. C. Mott
John A. Connolly	Francis A. Williams
Wiltsie F. Wolfe, *Pres.*	Stewart A. Jellett, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams	W. S. Hadaway, Jr.
Albert A. Cryer	Wm. McMannis
Wiltsie F. Wolfe, *Pres.*	Stewart A. Jellett, *Secy.*

1899

President	Henry Adams
1st Vice-President	D. M. Quay
2nd Vice-President	A. E. Kenrick
3rd Vice-President	Francis A. Williams
Treasurer	Judson A. Goodrich
Secretary	Wm. M. Mackay

Board of Managers

Chairman, Stewart A. Jellett

B. H. Carpenter	Wm. Kent
A. A. Cary	Wiltsie F. Wolfe
Henry Adams, *Pres.*	Wm. M. Mackay, *Secy.*

Council

Chairman, R. C. Carpenter

John Gormly	Wm. McMannis
W. S. Hadaway, Jr.	B. F. Stangland
Henry Adams, *Pres.*	Wm. M. Mackay, *Secy.*

1900

President....D. M. Quay
1st Vice-President....A. E. Kenrick
2nd Vice-President....Francis A. Williams
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Governors

Chairman, D. M. Quay

Wm. Kent, *Vice-Chm.* D. M. Nesbit
R. C. Carpenter C. B. J. Snyder
John Gormly Wm. M. Mackay, *Secy.*

1901

President....J. H. Kinealy
1st Vice-President....A. E. Kenrick
2nd Vice-President....Andrew Harvey
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Governors

Chairman, J. H. Kinealy

Wm. Kent, *Vice-Chm.* John Gormly
R. C. Carpenter C. B. J. Snyder
R. P. Bolton Wm. M. Mackay, *Secy.*

1902

President....A. E. Kenrick
1st Vice-President....Andrew Harvey
2nd Vice-President....Robert C. Clarkson
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Governors

Chairman, A. E. Kenrick

John Gormly, *Vice-Chm.* J. H. Kinealy
R. C. Carpenter C. B. J. Snyder
Wm. Kent Wm. M. Mackay, *Secy.*

1903

President....H. D. Crane
1st Vice-President....Wm. Kent
2nd Vice-President....R. P. Bolton
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Governors

Chairman, H. D. Crane

C. B. J. Snyder, *Vice-Chm.* A. E. Kenrick
R. C. Carpenter Geo. Mehring
John Gormly Wm. M. Mackay, *Secy.*

1904

President....Andrew Harvey
1st Vice-President....John Gormly
2nd Vice-President....Robert C. Clarkson
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, Andrew Harvey

John Gormly H. D. Crane
Robert C. Clarkson A. E. Kenrick
J. J. Blackmore C. B. J. Snyder
R. C. Carpenter Wm. M. Mackay, *Secy.*

1905

President....Wm. Kent
1st Vice-President....R. P. Bolton
2nd Vice-President....C. B. J. Snyder
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, Wm. Kent

R. P. Bolton James Mackay
C. B. J. Snyder B. F. Stangland
B. H. Carpenter J. C. F. Trachsel
A. B. Franklin Wm. M. Mackay, *Secy.*

1906

President....John Gormly
1st Vice-President....C. B. J. Snyder
2nd Vice-President....T. J. Waters
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, John Gormly

C. B. J. Snyder, *Vice-Chm.* James Mackay
R. C. Carpenter B. F. Stangland
Frank K. Chew T. J. Waters
A. B. Franklin Wm. M. Mackay, *Secy.*

1907

President....C. B. J. Snyder
1st Vice-President....James Mackay
2nd Vice-President....Wm. G. Snow
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, C. B. J. Snyder

James Mackay, *Vice-Chm.* Frank K. Chew
R. E. Atkinson A. B. Franklin
R. C. Carpenter Wm. G. Snow
Edmund F. Capron Wm. M. Mackay, *Secy.*

1908

President....James Mackay
1st Vice-President....Jas. D. Hoffman
2nd Vice-President....B. F. Stangland
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, James Mackay

Jas. D. Hoffman, *Vice-Chm.* John F. Hale
B. F. Stangland August Kehm
R. C. Carpenter C. B. J. Snyder
Frank K. Chew Wm. M. Mackay, *Secy.*

1909

President....Wm. G. Snow
1st Vice-President....August Kehm
2nd Vice-President....B. S. Harrison
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, Wm. G. Snow

August Kehm, *Vice-Chm.* Samuel R. Lewis
John R. Allen James Mackay
R. C. Carpenter B. F. Stangland
B. S. Harrison Wm. M. Mackay, *Secy.*

1910

President........Jas. D. Hoffman
1st Vice-President........R. P. Bolton
2nd Vice-President........Samuel R. Lewis
Treasurer........Ulysses G. Scollay
Secretary........Wm. M. Mackay

Board of Governors

Chairman, Jas. D. Hoffman

R. P. Bolton, *Vice-Chm.*
Geo. W. Barr
R. C. Carpenter
Judson A. Goodrich
John F. Hale
Samuel R. Lewis
James Mackay
Wm. M. Mackay, *Secy.*

1911

President........R. P. Bolton
1st Vice-President........John R. Allen
2nd Vice-President........A. B. Franklin
Treasurer........Ulysses G. Scollay
Secretary........Wm. W. Macon

Board of Governors

Chairman, R. P. Bolton

John R. Allen, *Vice-Chm.*
John T. Bradley
R. C. Carpenter
James H. Davis
A. B. Franklin
Jas. D. Hoffman
August Kehm
Wm. W. Macon, *Secy.*

1912

President........John R. Allen
1st Vice-President........John F. Hale
2nd Vice-President........Edmund F. Capron
Treasurer........James A. Donnelly
Secretary........Wm. W. Macon

Board of Governors

Chairman, John R. Allen

John F. Hale, *Vice-Chm.*
Edmund F. Capron
R. P. Bolton
Jas. D. Hoffman
Dwight D. Kimball
Samuel R. Lewis
Wm. M. Mackay
Wm. W. Macon, *Secy.*

1913

President........John F. Hale
1st Vice-President........A. B. Franklin
2nd Vice-President........Edmund F. Capron
Treasurer........James A. Donnelly
Secretary........Edwin A. Scott

Board of Governors

Chairman, John F. Hale

A. B. Franklin, *Vice-Chm.*
John R. Allen
Edmund F. Capron
R. P. Bolton
Frank T. Chapman
Ralph Collamore
James A. Donnelly
Dwight D. Kimball
Wm. W. Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President........Samuel R. Lewis
1st Vice-President........Edmund F. Capron
2nd Vice-President........Dwight D. Kimball
Treasurer........James A. Donnelly
Secretary........J. J. Blackmore

Council

Chairman, Samuel R. Lewis

E. F. Capron, *Vice-Chm.*
Dwight D. Kimball
John R. Allen
Frank T. Chapman
Frank I. Cooper
James A. Donnelly
John F. Hale
Harry M. Hart
Frank G. McCann
Wm. W. Macon
James M. Stannard
J. J. Blackmore, *Secy.*

1915

President........Dwight D. Kimball
1st Vice-President........Harry M. Hart
2nd Vice-President........Frank T. Chapman
Treasurer........Homer Addams
Secretary........J. J. Blackmore

Council

Chairman, Dwight D. Kimball

Harry M. Hart, *Vice-Chm.*
Homer Addams
Frank T. Chapman
Frank I. Cooper
E. Vernon Hill
Wm. M. Kingsbury
Samuel R. Lewis
Frank G. McCann
J. T. J. Mellon
Henry C. Meyer, Jr.
Arthur K. Ohmes
J. J. Blackmore, *Secy.*

1916

President........Harry M. Hart
1st Vice-President........Frank T. Chapman
2nd Vice-President........Arthur K. Ohmes
Treasurer........Homer Addams
Secretary........Casin W. Obert

Council

Chairman, Harry M. Hart

F. T. Chapman, *Vice-Chm.*
Homer Addams
Charles R. Bishop
Frank I. Cooper
Milton W. Franklin
E. Vernon Hill
Dwight D. Kimball
Henry C. Meyer, Jr.
Arthur K. Ohmes
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1917

President........J. Irvine Lyle
1st Vice-President........Arthur K. Ohmes
2nd Vice-President........Fred R. Still
Treasurer........Homer Addams
Secretary........Casin W. Obert

Council

Chairman, J. Irvine Lyle

A. K. Ohmes, *Vice-Chm.*
Homer Addams
Davis S. Boyden
Bert C. Davis
Milton W. Franklin
Charles A. Fuller
Harry M. Hart
E. Vernon Hill
James M. Stannard
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1918

President........Fred R. Still
1st Vice-President........Walter S. Timmis
2nd Vice-President........E. Vernon Hill
Treasurer........Homer Addams
Secretary........Casin W. Obert

Council

Chairman, Fred R. Still

W. S. Timmis, *Vice-Chm.*
Homer Addams
William H. Driscoll
Howard H. Fielding
H. P. Gant
C. W. Kimball
J. Irvine Lyle
E. Vernon Hill
Frank G. Phegley
Fred. W. Powers
Champlain L. Riley
Casin W. Obert, *Secy.*

1919

President........Walter S. Timmis
1st Vice-President........E. Vernon Hill
2nd Vice-President........Milton W. Franklin
Treasurer........Homer Addams
Secretary........Casin W. Obert

Council

Chairman, Walter S. Timmis

E. Vernon Hill, *Vice-Chm.*
Homer Addams
Howard H. Fielding
Milton W. Franklin
Harry E. Gerrish
George B. Nichols
Frank G. Phegley
Fred. W. Powers
Robt. W. Pryor, Jr.
Champlain L. Riley
Fred R. Still
Casin W. Obert, *Secy.*

1920

President .... E. Vernon Hill
1st Vice-President .... Champlain L. Riley
2nd Vice-President .... Jay R. McColl
Treasurer .... Homer Addams
Secretary .... Casin W. Obert

Council

Chairman, E. Vernon Hill

C. L. Riley, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Wm. H. Driscoll
A. C. Edgar
Alfred Kellogg
Jay R. McColl
George B. Nichols
Robt. W. Pryor, Jr.
W. S. Timmis
Perry West
Casin W. Obert, *Secy.*

1921

President .... Champlain L. Riley
1st Vice-President .... Jay R. McColl
2nd Vice-President .... H. P. Gant
Treasurer .... Homer Addams
Secretary .... Casin W. Obert

Council

Chairman, Champlain L. Riley

Jay R. McColl, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
H. P. Gant
E. S. Hallett
E. Vernon Hill
Alfred Kellogg
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1922

President .... Jay R. McColl
1st Vice-President .... H. P. Gant
2nd Vice-President .... Samuel E. Dibble
Treasurer .... Homer Addams
Secretary .... Casin W. Obert

Council

Chairman, Jay R. McColl

H. P. Gant, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
E. S. Hallett
L. A. Harding
E. E. McNair
H. J. Meyer
C. L. Riley
Perry West
Casin W. Obert, *Secy.*

1923

President .... H. P. Gant
1st Vice-President .... Homer Addams
2nd Vice-President .... E. E. McNair
Treasurer .... Wm. H. Driscoll
Secretary .... C. W. Obert

Council

Chairman, H. P. Gant

Homer Addams, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
Wm. H. Driscoll
E. S. Hallett
Alfred Kellogg
Thornton Lewis
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1924

President .... Homer Addams
1st Vice-President .... S. E. Dibble
2nd Vice-President .... William H. Driscoll
Treasurer .... Perry West
Secretary .... F. C. Houghten

Council

Chairman, Homer Addams

S. E. Dibble, *Vice-Chm.*
F. Paul Anderson
W. H. Carrier
J. A. Cutler
William H. Driscoll
H. P. Gant
W. E. Gillham
L. A. Harding
Alfred Kellogg
Thornton Lewis
Perry West
F. C. Houghten, *Secy.*

1925

President .... S. E. Dibble
1st Vice-President .... Wm. H. Driscoll
2nd Vice-President .... F. Paul Anderson
Treasurer .... Perry West
Secretary .... F. C. Houghten

Council

Chairman, S. E. Dibble

Wm. H. Driscoll, *Vice-Chm.*
Homer Addams
F. Paul Anderson
W. H. Carrier
J. A. Cutler
W. E. Gillham
W. T. Jones
Thornton Lewis
J. H. Walker
Perry West
A. C. Willard
F. C. Houghten, *Secy.*

1926

President .... W. H. Driscoll
1st Vice-President .... F. Paul Anderson
2nd Vice-President .... A. C. Willard
Treasurer .... W. E. Gillham
Secretary .... A. V. Hutchinson

Council

Chairman, W. H. Driscoll

F. Paul Anderson, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
W. E. Gillham
C. V. Haynes
W. T. Jones
E. B. Langenberg
Thornton Lewis
J. F. McIntire
A. C. Willard

1927

President .... F. Paul Anderson
1st Vice-President .... A. C. Willard
2nd Vice-President .... Thornton Lewis
Treasurer .... W. E. Gillham
Secretary .... A. V. Hutchinson

Council

Chairman, F. Paul Anderson

A. C. Willard, *Vice-Chm.*
H. H. Angus
W. H. Carrier
W. H. Driscoll
Roswell Farnham
H. H. Fielding
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
Thornton Lewis
J. F. McIntire
H. Lee Moore
F. B. Rowley

1928

President .... A. C. Willard
1st Vice-President .... Thornton Lewis
2nd Vice-President .... L. A. Harding
Treasurer .... W. E. Gillham
Secretary .... A. V. Hutchinson

Council

Chairman, A. C. Willard

Thornton Lewis, *Vice-Chm.*
F. Paul Anderson
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
J. F. McIntire
H. Lee Moore
F. B. Rowley

1929

President .... Thornton Lewis
1st Vice-President .... L. A. Harding
2nd Vice-President .... W. H. Carrier
Treasurer .... W. E. Gillham
Secretary .... A. V. Hutchinson
Technical Secretary .... P. D. Close

Council

Chairman, Thornton Lewis

L. A. Harding, *Vice-Chm.*
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
W. A. Rowe
F. B. Rowley
A. C. Willard

1930

President......L. A. Harding
1st Vice-President......W. H. Carrier
2nd Vice-President......F. B. Rowley
Treasurer......C. W. Farrar
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, L. A. Harding

W. H. Carrier, *Vice-Chm.*
H. H. Angus
D. S. Boyden
R. H. Carpenter
J. D. Cassell
N. W. Downes
Roswell Farnham
C. W. Farrar
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
Thornton Lewis
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1931

President......W. H. Carrier
1st Vice-President......F. B. Rowley
2nd Vice-President......W. T. Jones
Treasurer......F. D. Mensing
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, W. H. Carrier

F. B. Rowley, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
Roswell Farnham
E. H. Gurney
L. A. Harding
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
F. D. Mensing
W. A. Rowe

1932

President......F. B. Rowley
1st Vice-President......W. T. Jones
2nd Vice-President......C. V. Haynes
Treasurer......F. D. Mensing
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, F. B. Rowley

W. T. Jones, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
W. H. Carrier
John D. Cassell
E. O. Eastwood
Roswell Farnham
F. E. Giesecke
E. H. Gurney
C. V. Haynes
John Howatt
G. L. Larson
J. F. McIntire
F. D. Mensing
W. E. Stark

1933

President......W. T. Jones
1st Vice-President......C. V. Haynes
2nd Vice-President......John Howatt
Treasurer......D. S. Boyden
Secretary......A. V. Hutchinson

Council

Chairman, W. T. Jones

C. V. Haynes, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
R. Farnham
F. E. Giesecke
E. H. Gurney
John Howatt
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. W. Moon
F. B. Rowley
W. E. Stark

1934

President......C. V. Haynes
1st Vice-President......John Howatt
2nd Vice-President......G. L. Larson
Treasurer......D. S. Boyden
Secretary......A. V. Hutchinson

Council

Chairman, C. V. Haynes

John Howatt, *Vice-Chm.*
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
W. T. Jones
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. Walter Moon
O. W. Ott
W. A. Russell
W. E. Stark

1935

President......John Howatt
1st Vice-President......G. L. Larson
2nd Vice-President......D. S. Boyden
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson

Council

Chairman, John Howatt

G. L. Larson, *Vice-Chm.*
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
C. V. Haynes
J. F. McIntire
F. C. McIntosh
L. Walter Moon
A. J. Offner
O. W. Ott
W. A. Russell
W. E. Stark

1936

President......G. L. Larson
1st Vice-President......D. S. Boyden
2nd Vice-President......E. H. Gurney
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson

Council

Chairman, G. L. Larson

D. S. Boyden, *Vice-Chm.*
M. C. Beman
R. C. Bolsinger
Albert Buenger
S. H. Downs
W. L. Fleisher
F. E. Giesecke
E. H. Gurney
John Howatt
C. M. Humphreys
L. Walter Moon
J. F. McIntire
A. J. Offner
O. W. Ott
W. A. Russell
W. E. Stark

1937

President......D. S. Boyden
1st Vice-President......E. Holt Gurney
2nd Vice-President......J. F. McIntire
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson

Council

Chairman, D. S. Boyden

E. H. Gurney, *Vice-Chm.*
J. J. Aeberly
M. C. Beman
R. C. Bolsinger
Albert Buenger
S. H. Downs
E. O. Eastwood
W. L. Fleisher
F. E. Giesecke
C. M. Humphreys
G. L. Larson
W. A. Russell
W. E. Stark

LIST OF MEMBERS
Geographically Arranged

UNITED STATES
and
ISLAND TERRITORIES

ALABAMA

Birmingham
- Cone, W. F.
- Fried, H. V.
- Gause, H. C.
- Hardy, F. L.
- Lichty, C. P.
- Murphree, R. L.
- Trawick, J. G.

Mobile
- Kistler, M. L.

ARIZONA

Lowell
- Vinson, Neal L.

Phoenix
- Hummel, G. W.

ARKANSAS

Little Rock
- McCoy, C. E.

Pine Bluff
- Greer, W. R.

Siloam Springs
- Jones, C. R.

CALIFORNIA

Albany
- Kaup, E. O.

Bakersfield
- Baker, H. S.

Berkeley
- Bentley, C. E.
- Cherry, V. H.
- Foote, A. G.
- Raber, B. F.
- Woods, B. M.

Beverly Hills
- Theobald, A.

Culver City
- Owen, J. D.

Fullerton
- Miles, C. N.

Glendale—
- Moon, F. L.
- Orear, A. G.
- Storms, R. M.

Los Angeles—
- Anderson, C. S.
- Baender, F. G.
- Berglund, N. W.
- Blumenthal, M. I.
- Bullock, H. H.
- Cawby, E. L.
- Cline, E. A.
- Cooper, A. W.
- Cranston, W. E., Jr.
- Dickinson, T.
- Douglas, H. H.
- Downes, A. H.
- Ellingwood, E. L.
- English, H.
- Hajek, W. J.
- Hendrickson, H. M.
- Hess, A. J.
- Hill, F. M.
- Hogue, W. M.
- Hungerford, L.
- Kendall, E. H.
- Kennedy, M.
- Kilpatrick, W. S.
- Lauer, H. B.
- Leilich, R. K.
- Miller, G.
- Moriarty, J. M.
- Nelson, E. L.
- Ness, W. H. C.
- Ott, O. W.
- Park, J. F.
- Phillips R. E.
- Polderman, L. H.
- Schechter, J. E.
- Scofield, P. C.
- Wallace, K. S.
- Weidele, E. J.

Oakland—
- Cummings, G. J.
- Mears, L. A.

Orinda—
- Harrison, G. G.

Pacific Palisades—
- Finney, B.

Palo Alto—
- Johnson, O. W.

Pasadena—
- Gifford, R. L.

Piedmont—
- Gayner, J.

Sacramento—
- Freeman, J. C.

San Diego—
- Sadler, C. B.

San Francisco—
- Ames, C. S.
- Anaya, M.
- Bouey, A. J.
- Cochran, L. H.
- Cockins, W. W.
- Conrad, R.
- Cooley, E. C.
- Corrao, J.
- Feyge, H.
- Haley, H. S.
- Higdon, H. S.
- Hill, J. A.
- Holland, R. B.
- Hook, F. W.
- Hudson, R. A.
- Kindorf, H. L.
- Kooistra, J. F.
- Krueger, J. I.
- Leland, W. E.
- Marshall, T. A.
- O'Connor, G. P.
- Peterson, N. H.
- Simonson, G. M.
- Ward, E. B.
- Wayland, C. E.
- Wethered, W.
- White, T. J.

Santa Monica—
- Coghlan, S. F.

Sausalito—
- Howe, W. W.

South Pasadena—
- Warren, H. L.

West Los Angeles—
- Fabling, W. D.
- Lehmann, M.

COLORADO

Colorado Springs—
- Jardine, D. C.

Denver—
- Davis, A. F.
- McNevin, J. E.
- McQuaid, D. J.
- O'Rear, L. R.
- Pierce, E. D.
- Ward, O. G.

Fort Collins—
- Curtice, J. M.

CONNECTICUT

Bridgeport—
- Earle, F. E.
- Smak, J. R.

East Hartford—
- Pritchard, W. J.

Fairfield—
- Osborn, W. J.

Glenbrook—
- Wahrenbrock, O. K.

Greenwich—
- Jones, A. L.
- Opperman, E. F.

Hartford—
- Krintzman, H.

Middlebury—
- Lincoln, R. L.

New Britain—
- Leupold, H. W.

New Haven—
- Blakeley, H. J.
- Rodee, E. J.
- Seeley, L. E.
- Teasdale, L. A.
- Williams, G. S.
- Winslow, C.-E. A.

New London—
- Chapin, C. G.
- Forsberg, W.
- Hopson, W. T.

South Norwalk—
- Adams, H. E.
- Harvey, A. D.
- Jennings, I. C.
- Lyons, C. J.
- Mead, E. A.
- Wylie, H. M.

Stamford—
- Hoyt, L. W.
- Jehle, F.
- Jessup, B. H.

Torrington—
- Doster, A.

Wallingford—
- Burns, J. R.

Waterbury—
- Simpson, W. K.
- Stewart, C. W.

DELAWARE

Wilmington—
- Belt, N. O.
- Gawthrop, F. H.
- Hayman, A. E., Jr.
- Kershaw, M. G.
- Lownsbery, B. F.
- Ponsell, F. I.
- Robinson, G. L.
- Schoenijahn, R. P.
- Shepherd, C. B.